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DIESEL-FIRED SELF-PUMPING WATER HEATER

By
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Advanced Mechanical Technology, Inc.
Newton, MA 02158

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The object of this project was to study the feasibility of pumping and heating water by sustained ocillatory vaporization and condensation in a fired heat exchanger. Portable field liquid fueled water heaters would facilitate heating water for sanitation, personal hygiene, food service, laundry, equipment maintenance and decontamination presently, available only from larger, less portable, motorized pumping units.

The technical tasks consisted of: development of an analytical model, operation of proof-of-principal prototypes, determination of the thermodynamic and mechanical relationships to evaluate operating range and control characteristics. Four successive pump models were analysed and tested. The final analytical model gave reasonable agreement with the experimental results, indicating that the actual pumping effect was an order of magnitude lower than originally anticipated.

It was concluded that a thermally-activated self-pumping water heater based on the proposed principle is not feasible.

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PREFACE

The object of this project was to study the feasibility of pumping and heating water by sustained oscillatory vaporization and condensation in a fired heat exchanger. Portable field liquid-fueled self-pumping water heaters would facilitate heating water for sanitation, personal hygiene, food service, laundry, equipment maintenance and decontamination presently available only from larger, less portable, motorized pumping units.

The technical scope consisted of development of an analytical model of the thermodynamics and dynamics, operation of gas-fired proof-of-principle prototypes, and determination of the thermodynamic and mechanical relationships to evaluate operating range and control characteristics. If the promising results were obtained, the next step would be development of a practical liquid-fueled self-pumping water heater.

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DIESEL-FIRED SELF-PUMPING WATER HEATER

1.0 EXECUTIVE SUMMARY

This report describes the research and development of a diesel-fired self-pumping water heater conducted as Phase I of a Department of Defense Small Business Innovation Research (SBIR) project under U.S. Army contract number DAAK60-93-C-0035.

The object of the project is to investigate the feasibility of pumping and heating water by means of intermittent vaporization in a fired heat exchanger. Demonstration of the feasibility of a self-pumping water heater would lead the way to development of readily portable liquid-fueled water heaters suitable for field sanitation purposes. Such water heaters would greatly facilitate heating of water for personal hygiene, food service, and laundry; plus, they would provide pressurized hot water for equipment maintenance and decontamination that is presently available only from larger, less portable, motorized pumping units.

The technical tasks to establish feasibility consisted of:

1. Development of an analytical model of the fluid dynamics and thermal dynamics of intermittent vaporization within a tubular vapor generator.

2. Operation of a proof-of-principle prototype water heater incorporating the self-pumping feature.

3. Determination of the relationships between heat input, pressure, flow, temperature and operating frequency of the device, in order to evaluate its operating range and control characteristics.

An analytical model was developed and subsequently revised to reflect successive modifications in the proof-of-principle device. An experimental apparatus was built to demonstrate the thermally-actuated pumping process and to enable measurement of its performance. Initial attempts to obtain sustained self-oscillations were unsuccessful. The apparatus was modified in stages to isolate the effects in an effort to gain an understanding of the governing phenomena.

Eventually, four separate pump models were analyzed and tested, and self-sustained oscillations were ultimately achieved. However, the pressure and flow amplitudes were well below the levels predicted by the analytical model. The final revision of the analytical model which accounted for the moisture content of the driving steam gave reasonable agreement with the experimental results, indicating that the actual pumping effect was an order of magnitude lower than originally anticipated.

results, indicating that the actual pumping effect was an order of magnitude lower than originally anticipated.

As a result of this research, it is concluded that a thermally-activated self-pumping water heater based on the proposed principle will not be feasible.

2.0 WORKSTATEMENT

The following is the statement of work for the project:

In order to evaluate the feasibility of the self-pumping water heater, the Phase 1 technical approach will be to first formulate a model of the processes involved in the transient heating/cooling of the water/steam, coupled with the dynamics of the pulsating water flow. Next, a proof-of-principle prototype boiler will be constructed and tested to validate the analytical model and to demonstrate the performance of the system.

The Phase 1 work plan will be:

Task 1 - Analytical Model
Develop an analytical model describing: (a) the heat transfer and thermodynamics of the boiler, including the transient heating and cooling during filling and emptying of the boiler; (b) the thermodynamics of mixing of the steam with the propelled water; (c) the dynamics of the suction and supply water columns. The objective of the model will be to relate the system geometry and heat transfer to the pressure, flow and temperature of the pumped water.

The heat exchanger will be modelled as a lumped-parameter system. Steady-state heat transfer correlations will be used to model the heat transfer to/from the water/steam. A simple first-order transient heat transfer model of the boiler tubing will be used to model the dynamics of the wall temperature. A one-dimensional transient energy, momentum, and continuity model of the water/steam will be used to model its dynamics.

Following formulation of the model, it will be used to simulate the operation of the self-pumping water heater to investigate relationships between pressure, flow and temperature of the water. Subsequently, the model will be used in the design of the proof-of-principle water heater.

Task 2 - Prototype Design
The analytical model will be used to design the heat exchanger and fluid piping of a 200,000 Btu/h self-pumping water heater. In order to focus on the crucial issues of

system dynamics, the heat exchanger will be designed to be heated by a gas burner, which will be simpler to construct and operate than the vaporizing diesel burner envisaged for the eventual unit.

The inlet and outlet piping will be designed so that its length and/or diameter can be readily changed order to vary its dynamics. A check valve will be used at the inlet of the boiler. At the outlet of the boiler, a solenoid valve will be used initially instead of a pressure-actuated poppet valve in order that its opening and closing can be manipulated. The boiler and piping will be instrumented with pressure, temperature, and flow sensors to be recorded dynamically.

Task 3 - Fabricate Boiler

A boiler coil will be fabricated out of commercially available finned tubing. The coil will be instrumented with thermocouples, and static and dynamic pressure probes for pressure and flow measurement. The pressure probes will be connected to solid-state pressure transducers for recording on a multi-channel strip chart recorder. The boiler coil will be fitted with a 200,000 Btu/h gas-fired power burner.

The test rig will consist of the gas-fired boiler coil and suction/supply piping. Water will be drawn from an open barrel and discharged into a weigh tank for flow measurement.

Task 4 - Test Boiler

Tests will be conducted over a range of firing rates from about 50,000 - 200,000 Btu/h. The water temperature rise will be varied over a range from about 50 - 150 °F by throttling the water flow rate. The pressure, temperature, and flow characteristics of the boiler will be mapped and compared to the predictions of the analytical model.

Task 5 - Phase 1 Report

A final report will be prepared describing the formulation of the analytical model and its predictions of the performance of the self-pumping water heater, the design of the proof-of-principle water heater, and its test results. Conclusions regarding the test results and the feasibility of the proposed water heater will be presented, and recommendations will be made regarding the technical approach to a practical diesel-fired self-pumping water heater.

3.0 BACKGROUND

The Army has numerous applications in the field for a portable source of hot water that does not require any external source of power. Presently, such needs are met by fired water heaters that heat a tank of water, such as the M67 liquid fuel fired immersion heater. Such batch-type heaters are slow and inefficient, and have no provision for filling the tank from its source of cold water, nor for supplying a pressurized flow of hot water to a remote point of use. When a continuous source of pressurized hot water is required, electrical or engine-driven water pumps must be utilized. This greatly limits the ease of portability and restricts access of forward field units to pressurize hot water for applications such as cooking, laundry, showers, decontamination, and equipment maintenance.

A portable water heater for forward field use ideally would require no external source of power other than standard diesel fuel; would provide continuous pumping from a remote source such as a pond or stream to the point of use; would be capable of modulating the temperature, pressure, and flow of hot water; would be compact and lightweight for easy portability; and would be simple and rugged for easy operation and maintenance.

While detailed specifications would depend on the particular application, typical specifications for the range of applications listed above would be:

Flow rate	2 - 5 GPM
Temperature rise	50 - 150°F
Pressure rise	20 - 200 psi
Heat input	75 - 200 MBTUH
Dry weight	50 - 100 lb
Volume	1 - 2 cu-ft
Fuel	gasoline - diesel

4.0 TECHNICAL APPROACH

The technical approach investigated in this project is to convert the thermal input of the water heater directly to pumping effort by vaporizing some of the water and using the steam to pressurize the water. Such direct-acting steam pumps tend to be inefficient, but since the water is to be heated anyway in the present application, this is no disadvantage so long as the rejected heat is used to heat the water.

The proposed self-pumping water heater incorporates three aspects that provide rapid vaporization and displacement of the water:

1. Water is preheated regeneratively by entering the boiler through tubing that has been heated by previously exiting hot water/steam mixture.

2. Instead of allowing the boiler vessel to cool below saturation temperature before allowing the fresh charge of water to enter, water is injected into the boiler at high velocity so that the water is in place before very much steam has been generated. Use of sufficiently ductile boiler tubing, combined with small temperature excursions, should preclude problems with thermal fatigue.

3. Filling of the boiler is accomplished rapidly and automatically by employing the inertia of the suction and supply lines to create overexpansion and condensation of the steam following the pressurization phase. This creates a partial vacuum in the boiler, which draws in the fresh charge of water.

The proposed water heater consists of a compact, coil-type heat exchanger fired by an atmospheric, vaporizing-type liquid fuel burner, as illustrated in Figure 1. The burner uses a pressurized fuel tank to supply fuel under pressure to a vaporizer, which is heated by the burner. Vaporized fuel is directed to a nozzle which produces a fuel vapor jet that entrains air for combustion. The fuel vapor and air mix thoroughly in a mixer tube which delivers the combustible mixture to the burner head. A pressure-atomizing preheater torch is used to preheat the vaporizer and ignite the main burner for start-up. This is a similar approach to that used by the offeror in the diesel fired cookstove developed under Army Contract No. DAAK60-87-C-0016.

The heat exchanger coil is open at only its top end, which is connected to the water intake/outlet piping. The closed end, which is at the bottom of the heat exchanger, contains a valve which is opened prior to start-up to permit filling the heat exchanger coil with water.

The pumping is accomplished by evaporating a small portion of the water in bottom of the heat exchanger. As the pressure rises, the steam pushes the water away from the closed end of the heat exchanger. As the steam reaches the in colder sections of the heat exchanger, it condenses. The momentum of the escaping water and steam causes overexpansion of the remaining steam at the bottom of the heat exchanger, creating a partial vacuum which draws in cold water through the inlet check valve. The momentum of the inrushing water draws it into the steam space, where it condenses/compresses most of the remaining steam, thereby refilling the boiler. After a delay period during which the water in the base of the boiler coil is heated and evaporated, the cycle begins again.

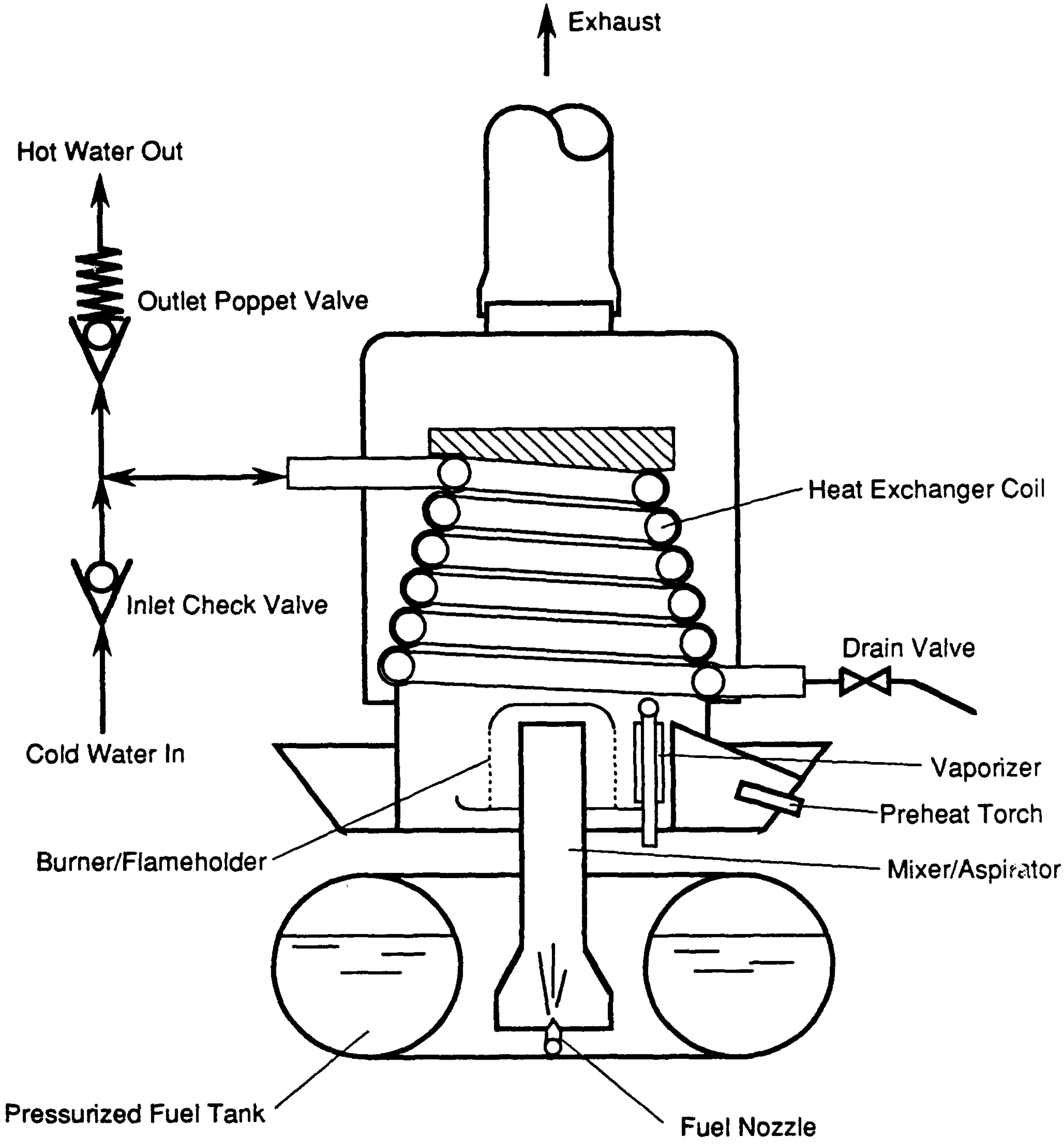


Figure 1. Original Self Pumping Water Heater Concept

5.0 ANALYTICAL AND EXPERIMENTAL DEVELOPMENT

The key element of the self-powered water heater is the process by which steam generated in the boiler draws in water and expels it at a higher pressure. Accordingly, this steam-driven pump was the focus of the feasibility investigation.

The process requires the steam to undergo a cycle in which the average pressure during its expansion phase, when it expelling the water, is greater than the pressure during its compression phase, when it is drawing in water from the supply reservoir. The proposed concept would use the water contained in the tubing of the boiler as a "liquid piston", displacing the steam between hotter and colder regions of tubing in order to affect its pressure. Furthermore, the mass of this liquid piston would provide the inertia necessary to sustain the oscillatory pumping.

Four substantially different analytical models were developed during the course of this project. Three of these models referred to different pumping configurations. In order to retain continuity between the analytical and experimental investigations, the analytical model and experimental results are described separately below for each version. The analytical models are described in general terms below; the reader is referred to Appendix A for the details of the models.

5.1 MK-I Pump

5.1.1 Analytical Model

The MK-I initial approach and its physical model is illustrated in Figure 2. The hydraulic circuit consists of the <u>inlet tube</u>, which is fed by the supply reservoir and connects to the junction of the <u>outlet tube</u> and the <u>regenerator</u>. The regenerator leads to the top outlet of the <u>boiler</u>, which is a tubular coil closed at its bottom end. Check valves are contained in the inlet and outlet tubes to control the direction of flow.

The intended operation is as follows: When the boiler is heated, steam is formed and propels the water column in the regenerator away from the boiler and out through the outlet tube. As the steam expands into the regenerator, it contacts the cooler walls of the regenerator and condenses, thereby reducing its pressure. The inertia of the liquid piston allows it to continue its outward motion while the steam pressure drops below atmospheric. The sub-atmospheric pressure draws water in through the inlet tube. Eventually, the low steam pressure causes the flow in the regenerator to reverse its direction, thereby compressing the steam and simultaneously reducing the contact

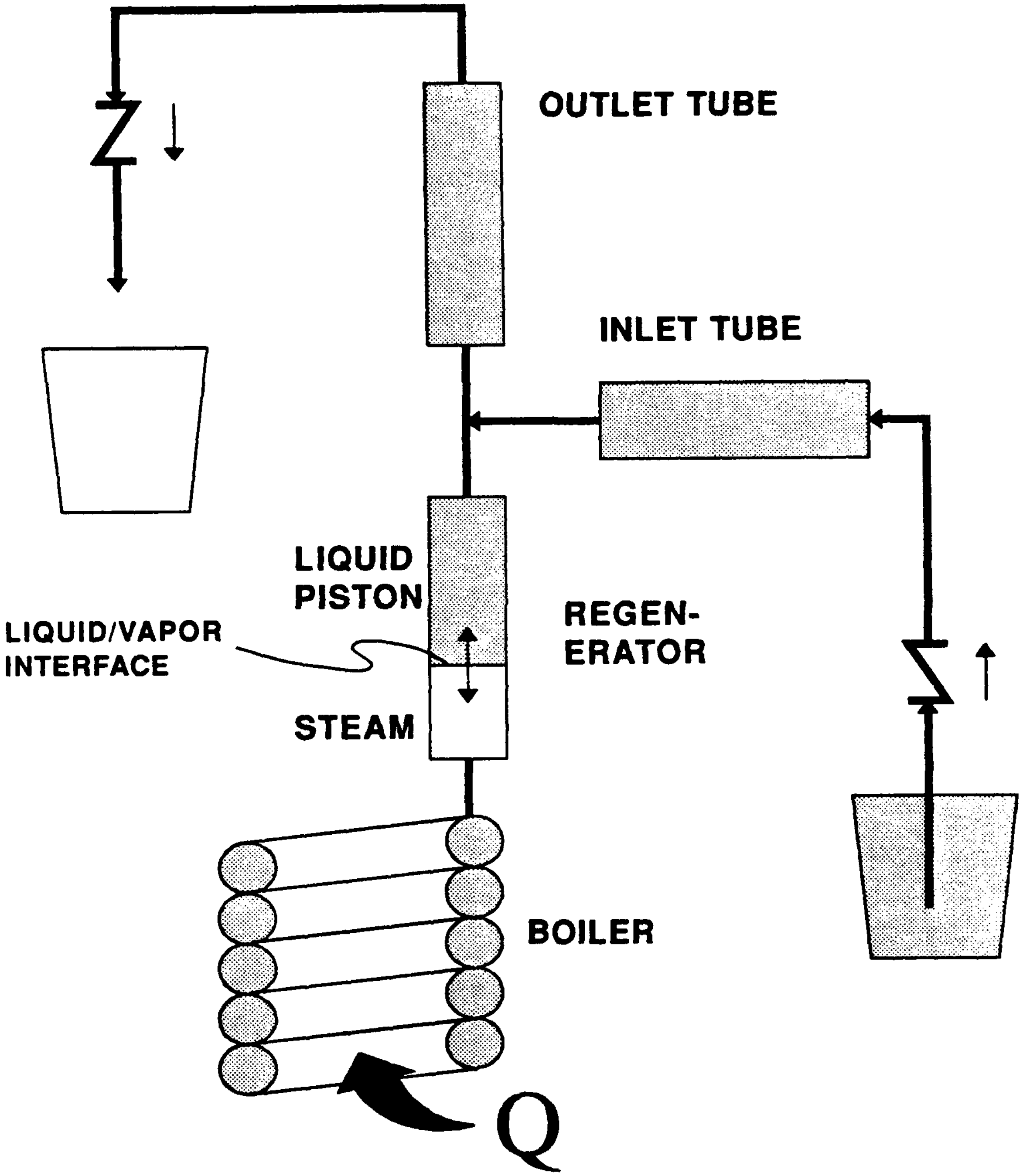


Figure 2. MK-I Pump Model

area of the steam with the cool walls. The continual evaporation in the boiler combined with the reduction in condenser area increases the amount of steam, leading to a rise in steam pressure. When the steam pressure rises above atmospheric, the incoming water column is decelerated, eventually reversing its direction, and the cycle repeats.

The purpose of the regenerator is to heat the water flowing toward the boiler, and to extract heat from the water flowing away from the boiler by storing heat in its wall. In the steady state, a temperature distribution will be established along the regenerator that will depend on the heat transferred by the condensing steam, as well as the net heat conveyed to the water.

The analytical model considered the dynamics of the separate columns of water in the inlet and outlet tubes and the regenerator. The model treated these as lumped masses, considering the continuity and momentum equations. Frictional losses were calculated using steady-state, fully-developed friction factors evaluated at the instantaneous Reynolds numbers. The inertia of the steam was considered to be negligible. Wall temperature distributions and heat transfer coefficients were assumed for the regenerator and the boiler. These determined the heat transfer to/from the steam and the water. (It was intended that the temperature distributions and heat transfer coefficients would ultimately be determined analytically; however, as will be seen below, this need did not materialize.)

The model assumed that there would always be sufficient residual water remaining in the boiler to be evaporated into steam. The rate of evaporation or condensation was determined by the heat transfer (through latent heat of vaporization). The net rate of evaporation determined the total mass of steam in the system. The volume of steam contained within the boiler and condensing portion of the regenerator was determined by the position of the liquid/vapor interface in the regenerator; i.e., through the dynamics of the "liquid piston". Knowledge of the mass and volume of steam gave the specific volume of the steam.

A key simplifying assumption at this point was to neglect the moisture in the steam, or in other words to assume dry, saturated steam. The rationale was that the energy change of the moisture would be small compared to the energy of the steam and liquid piston, and that such moisture could be considered as part of the liquid piston. This enabled the model to treat the steam as dry steam (100% quality), and to use the properties of dry saturated steam. Since the specific volume of dry saturated steam is a single-valued function of temperature (or pressure), this enabled the saturation temperature and pressure to be determined from knowledge of the specific volume. The resulting steam pressure became the driving force for the liquid-piston dynamics.

A Lotus 123 program was written (see Appendix) to simulate operation over a portion of a cycle using a time increment that ranged from 0.001 s to 0.005 s. The state points were stored after every 24 time increments (0.024 - 0.06 s), and up to 30 sets of state points (0.72 - 1.8 s) could be plotted to show all or part of a cycle.

This model indicated very poor pumping effect, and in order to obtain any appreciable amplitude it was necessary to eliminate viscous friction from the model. A typical output is shown in Figures 3 and 4. It is evident the pressure during the expansion stroke is only slightly greater than that during the compression stroke. Thus, although there is substantial pressure amplitude and stroke, the work per cycle is small since the pressure and displacement are almost in phase. To obtain greater work per cycle, the pressure must lag the displacement more.

It was found that the phase lag could be increased by reducing the boiling heat transfer coefficient or surface area, which caused the steam generation rate to lag the displacement by a greater amount. At this point the assumption that the boiling rate was determined by a constant wall temperature and heat transfer coefficient (and time-varying saturation temperature) was re-examined. Due to the long period of a cycle (>1 s), a rather massive wall would be required if its temperature was to remain constant. If instead the boiler wall was relatively thin and exposed to a high temperature burner flame, the assumption of constant heat flux would be a better approximation. Consequently, the constant boiler wall temperature model was discarded, and replaced with a constant heat flux model.

This was a propitious change, since it resulted in a larger pressure lag, which resulted in a "fatter" indicator diagram. The much larger work per cycle proved to be more than enough to overcome frictional losses and still provide a large pumping amplitude. A typical output is illustrated in Figures 5 and 6. Boiler pressure amplitude is +22/-9 psig, and displacement is about 3 ft at 0.7 Hz, corresponding to a flow of about 3 GPM. On the basis of this model, the initial proof-of-principle apparatus was built.

5.1.2 Experimental Results

The MK-I analytical model was used to generate a preliminary design for the proof-of-principle prototype. A schematic of the system is shown in Figure 7. Inlet, outlet, and regenerator tubing is 3/4" OD refrigeration tubing. A gas-fired, stainless-steel, 0.5" ID coil-type water heater was converted to a boiler. A baffle in the combustion chamber permits the heated length to be adjusted from approximately 9 feet up to 18 feet of tubing. The baffle was set at the 9-foot length. The gas input can be varied from about 25,000 Btu/h up to 50,000 Btu/h. Flow is measured with a 50 gallon weigh tank with resolution to 0.25 lb.

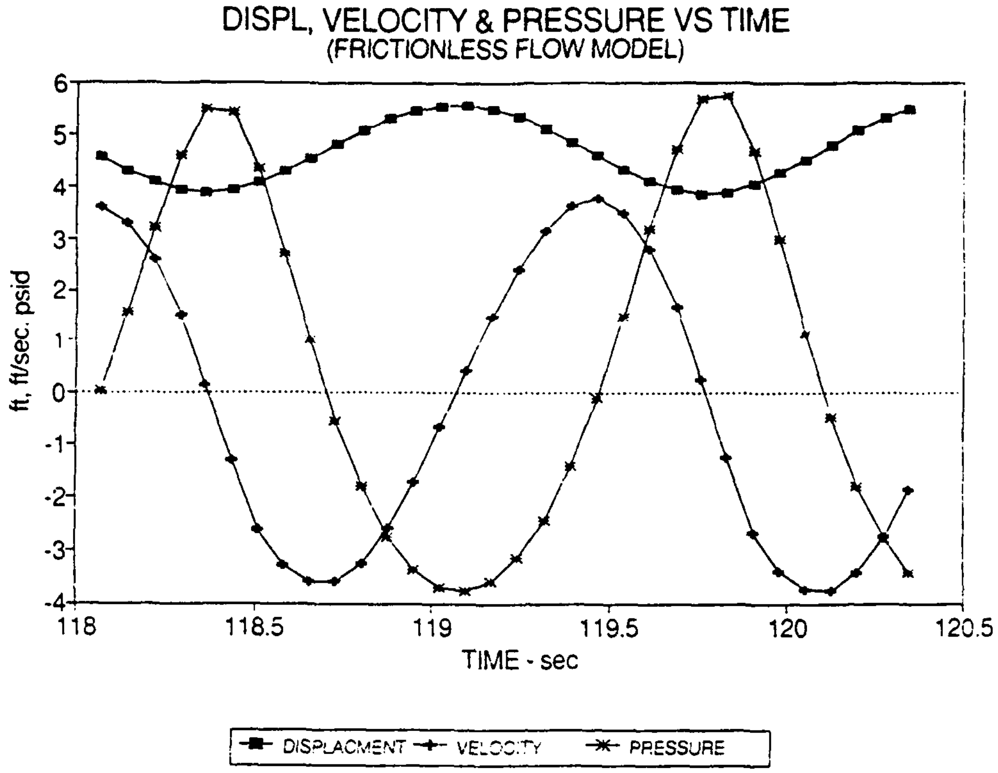


Figure 3. MK-I Cycle (Frictionless Model)

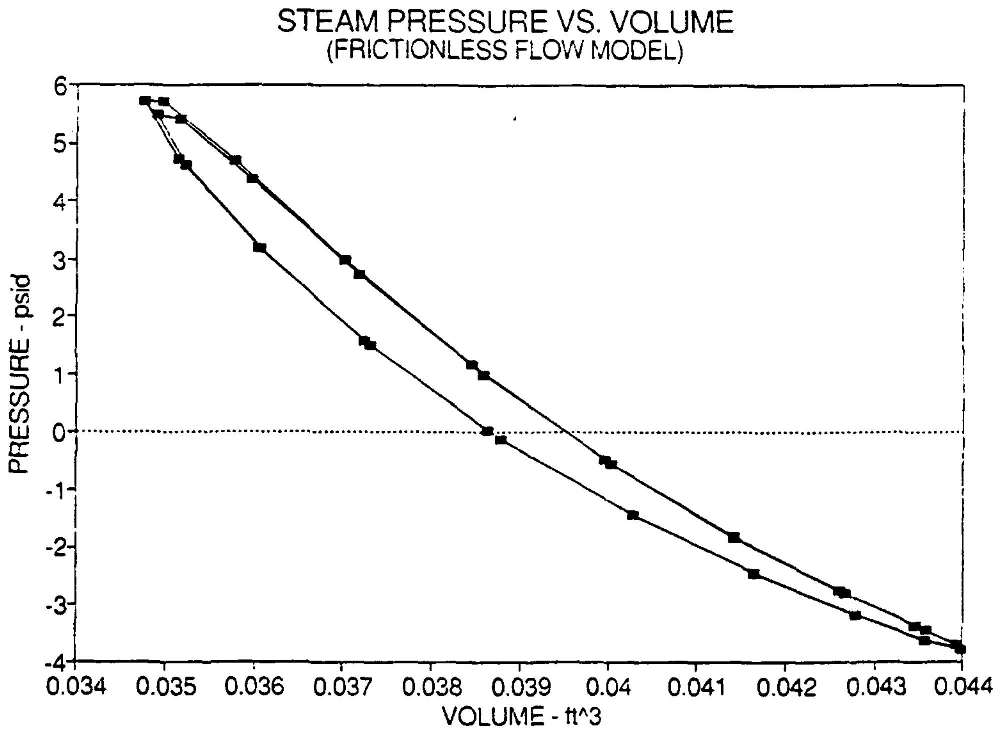


Figure 4. MK-I Indicator Diagram (Frictionless Model)

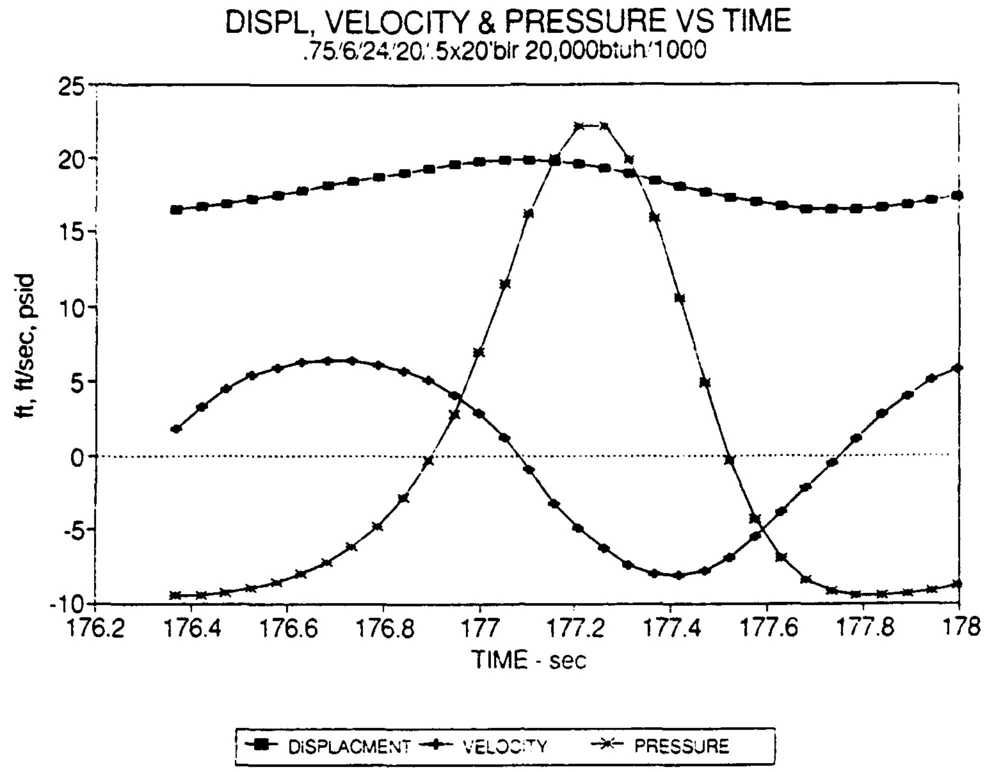


Figure 5. MK-I Cycle (Constant Heat Flux)

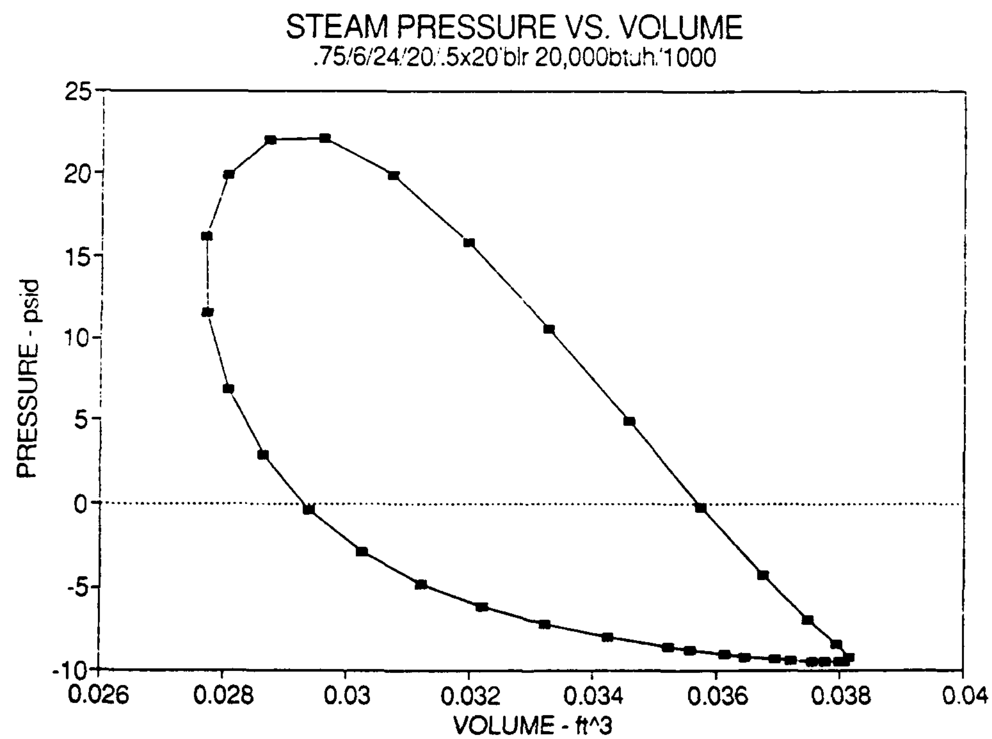


Figure 6. MK-I Indicator Diagram (Constant Heat Flux)

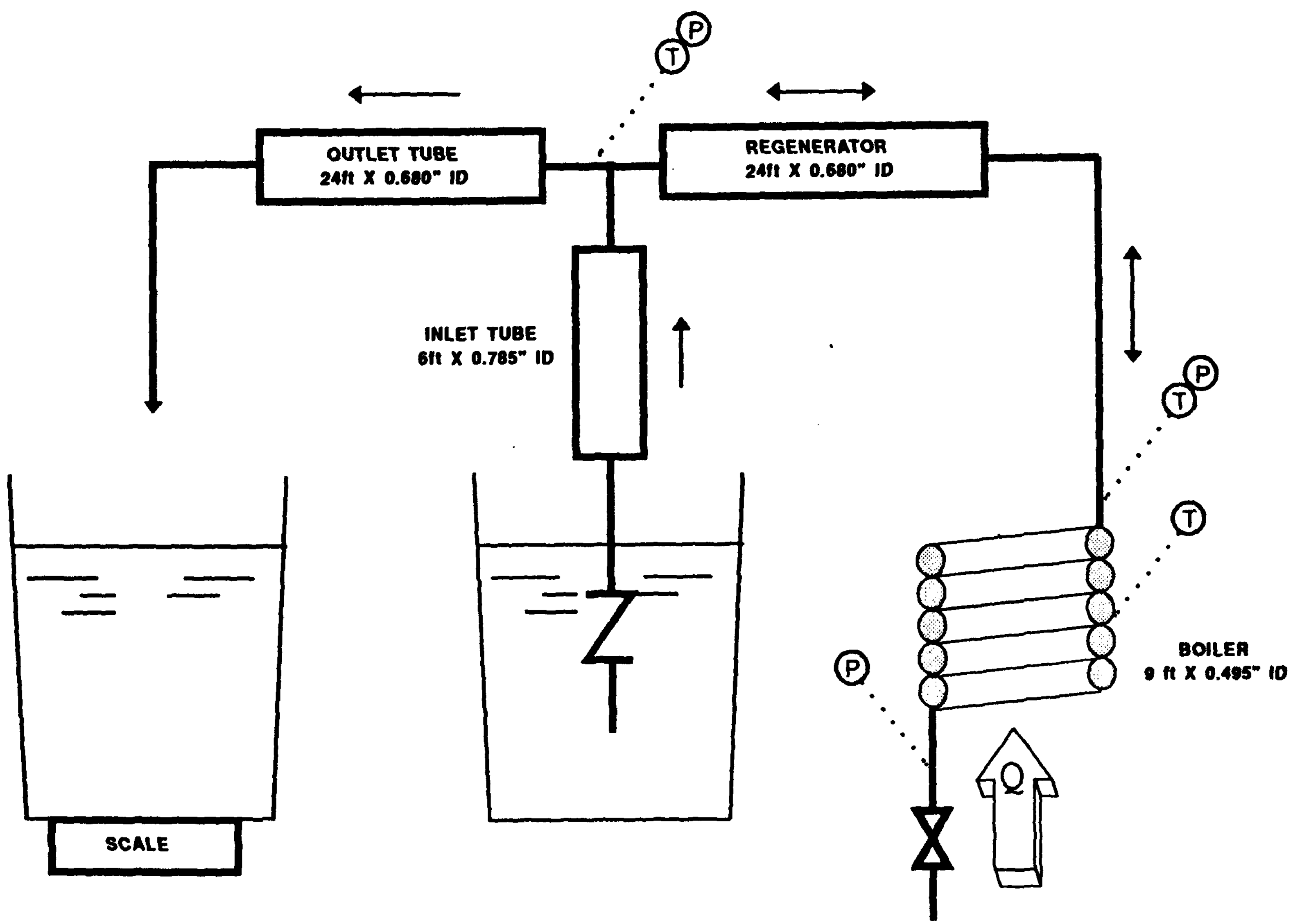


Figure 7. Initial MK-I Test Setup

Temperature instrumentation consists of thermocouples mounted on the outer wall of the boiler and on the wall of the copper tubing at the boiler outlet and between the regenerator and the outlet tube. Pressures are measured by two pressure transducers located at either two of three locations: at the dead end of the boiler, at the boiler outlet, and between the regenerator and the outlet tube. The MK-I test conditions are listed in Table 1.

Table 1. Initial MK-I Test Set-up

Inlet length	6 ft
Inlet ID	0.680 in
Outlet length	24 ft
Outlet ID	0.680 in
Regenerator length	24 ft
Regenerator ID	0.680 in
Boiler length	9 ft
Boiler ID	0.495 in
Maximum boiler heat input	50,000 Btu/h
Minimum boiler heat input	25,000 Btu/h

Data was recorded digitally on a data acquisition system. Water flow was measured by weighing the outflow into a barrel mounted on a platform scale. Gas input to the gas-fired boiler was measured by timing gas consumption on a gas meter.

Testing began with the configuration shown in Figure 7. The system was filled by connecting the house supply through the valve at the base of the boiler, and the inlet tube was primed with a hose. The burner firing rate was set at 40,000 Btu/h. When the burner was fired, the boiler tubing would heat until steam would be formed and the water would be expelled from the outlet tube, followed by surge of steam. Thereafter, the boiler would continue to heat, but there would be negligible additional expulsion of water and none of the expected oscillatory flow. The burner was allowed to fire until the boiler tubing reached 500°F.

Aside from the fact that no oscillations were observed, the main concern was that steam was issuing from the outlet tube instead of condensing in the regenerator. The assumption in the analytical model was that the cyclical heat of condensation would be stored in the wall of the regenerator tubing, which would be subsequently cooled by the incoming water column. The details of the thermal transient were not analyzed. However, an approximate calculation showed that the tube wall had a thermal time constant on the order of 0.25 s. Since the expected period of oscillation was longer than one second, it was clear that there was too little thermal inertia to condense the steam. An additional problem was that, unless oscillatory flow was produced, there would be no exchange of water in the regenerator to cool the regenerator walls.

To overcome this problem, the regenerator was placed in a water bath. This solved the problem of not condensing the steam, but there were still no sustained oscillations. However, there was an occasional intake/exhaust pulse after the initial expulsion of water.

It was observed that once the initial disturbance died down, there was no further change in pressure, since the outlet was open to atmosphere. To remedy this, a manual valve (and relief valve) was placed at the outlet. The outlet valve was manipulated in various ways to attempt to stimulate oscillations, with little success. Next, a check valve was placed at the outlet. This resulted in some transient oscillations, which were not sustained.

The modified configuration is illustrated in Figure 8. A rubber hose was added at the base of the boiler to act as an accumulator to reduce water hammer. The copper regenerator tubing is in a water bath. A check valve and ball valve were added to the outlet, but the ball valve was left open.

A typical test result is illustrated in Figure 9. As the boiler heats past 200°F, there is a period of rapid pressure oscillation followed by a period of lower frequency, lower amplitude oscillations. The steady increase in boiler temperature is interrupted by a sudden decline, which is the result of manually feeding some water into the boiler. Thereafter the temperature continues to rise to the point that the burner is shut off when the temperature reaches 500°F. The temperature continues to rise past that point, peaking at about 700°F, at which time the oscillations die down.

The period of these oscillations is approximately 2.5 seconds. This compares with a period of 1.1 - 1.6 seconds predicted by the MK-I analytical model. The total flow during the period of the test was about 3 gallons. Pressure amplitude is generally below 5 psi, versus up to +70/-10 psi predicted by the model.

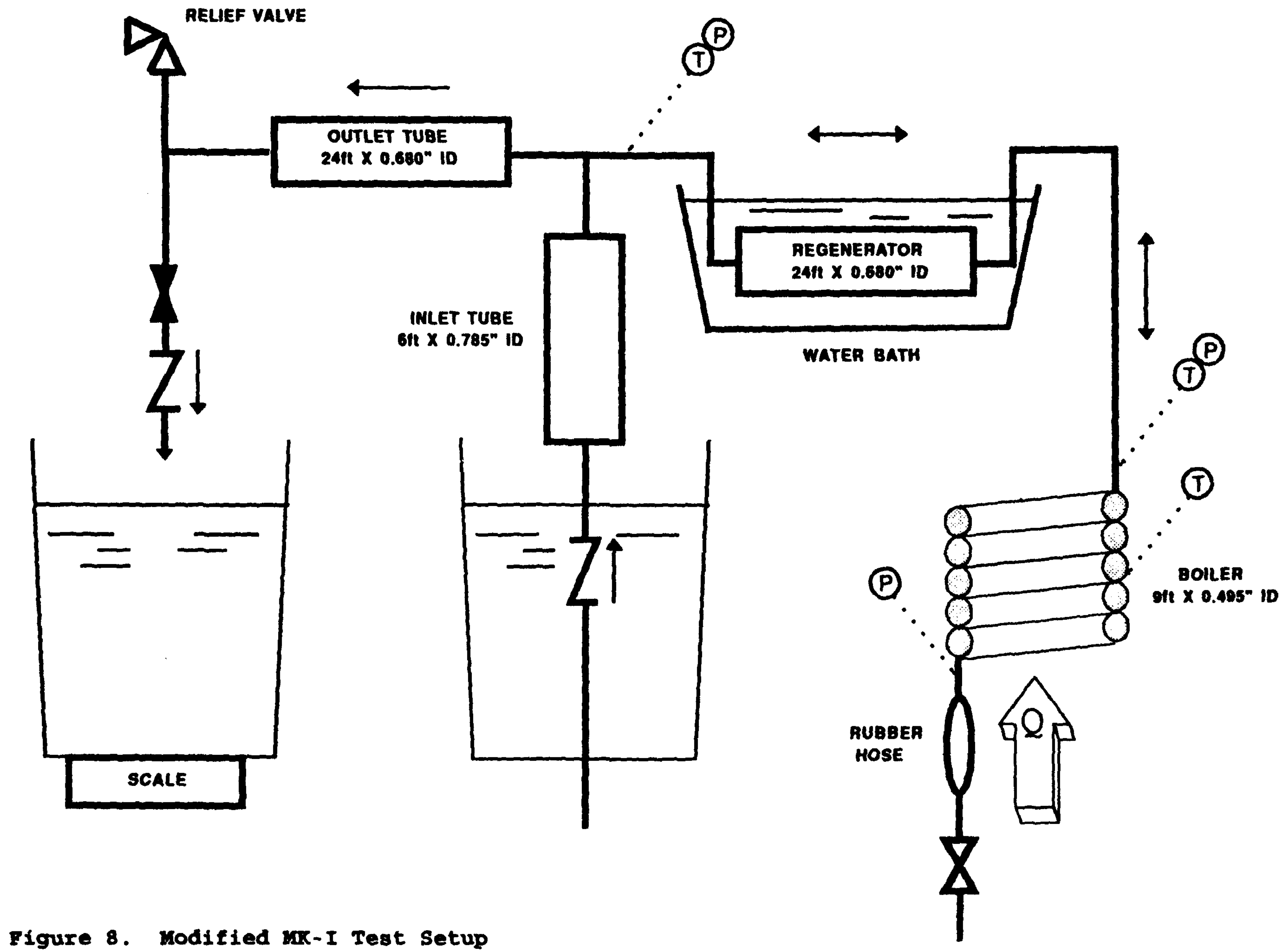


Figure 8. Modified MK-I Test Setup

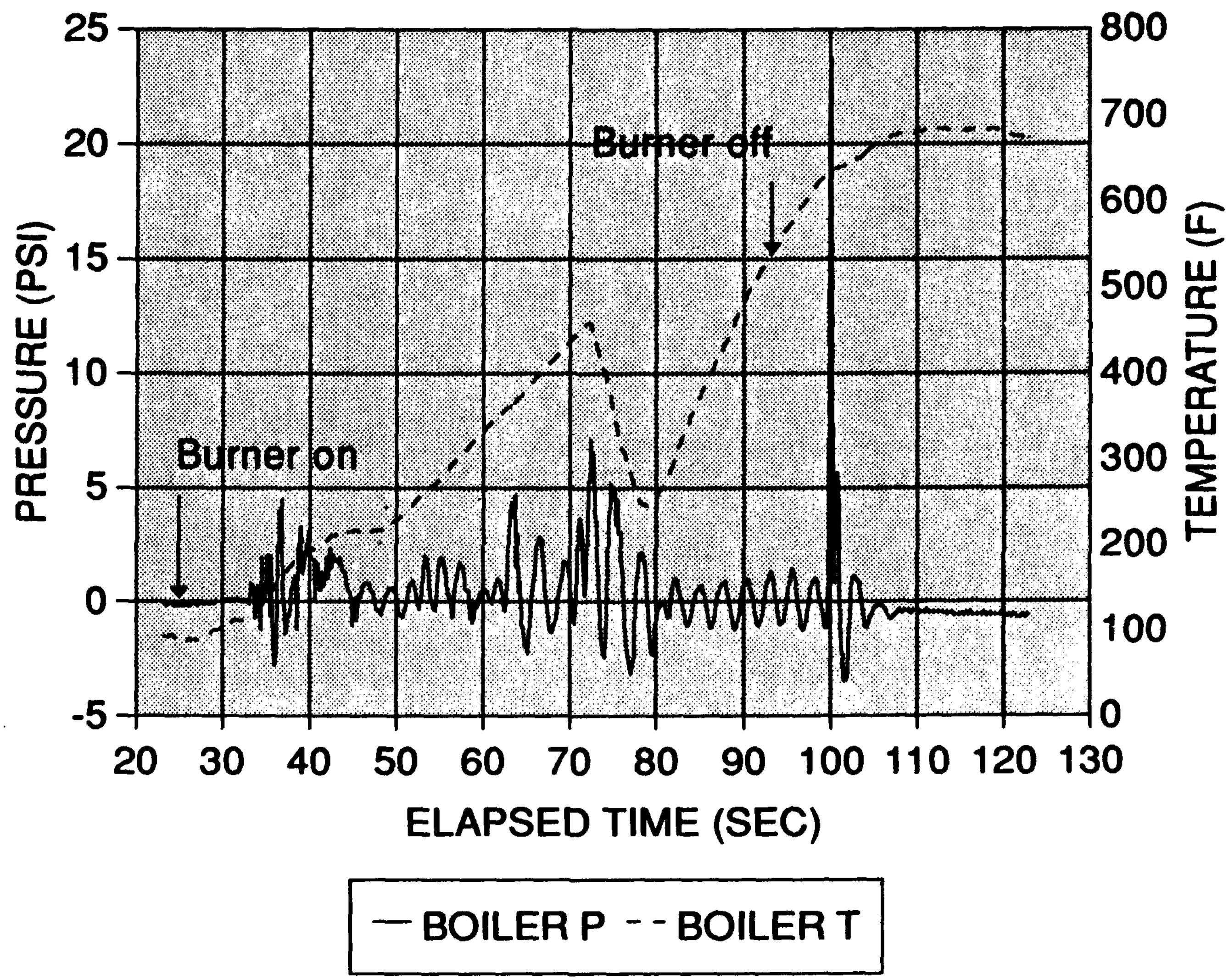


Figure 9. MK-I Test Results

Assessment of the test results obtained with the MK-I configuration suggested that even with the regenerator tube placed in a water bath, condensation of steam in the regenerator was occurring too slowly to sustain the oscillations. While the condensation heat transfer coefficient (HTC) inside the tube was estimated to be on the order of 2000 Btu/hr-ft^2-°F, the outside HTC due to natural convection of the water bath was only about 100 - 150 Btu/hr-ft^2-°F. Thus, the outside HTC was limiting the overall HTC to about 100 Btu/hr-ft^2-°F.

5.2 MK-II Pump

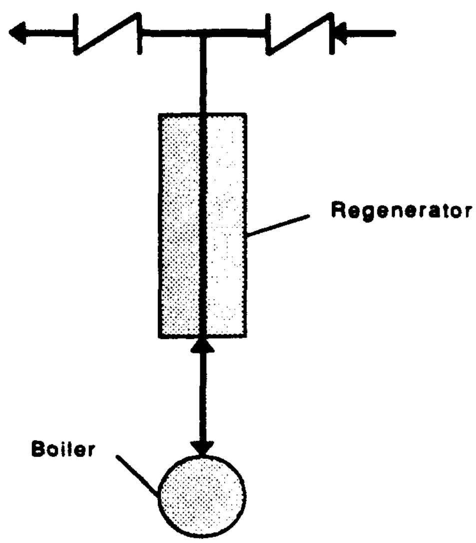
One of the findings of the MK-I analysis was that a high condensation rate was necessary for vigorous pumping. Without external cooling and with a low thermal mass, the regenerator produced a low condensation rate. For this reason, it was decided to replace the periodic-flow regenerator with a tube-in-tube heat exchanger. The inner tube would operate as a condenser/regenerator as before, but would be cooled by water flowing in the outer annulus. The water in the outer annulus would increase the effective heat capacity of the regenerator, while its velocity would provide a high heat transfer coefficient. The source of the cooling water could be either from the low-pressure supply or the higher pressure outlet flow. Thus the outer annulus would become one of the inlet/outlet liquid columns.

In the MK-I concept containing the single-tube reversing-flow regenerator, the boiler was "dead-ended". That is, flow entered the boiler from the regenerator and returned from the boiler through the same path, as illustrated in Figure 10(a). With the replacement of the single-tube regenerator by a tube-in-tube heat exchanger having two separate flow passages, the flow leaving the boiler traverses the regenerator through a different path, as illustrated in Figure 10(b).

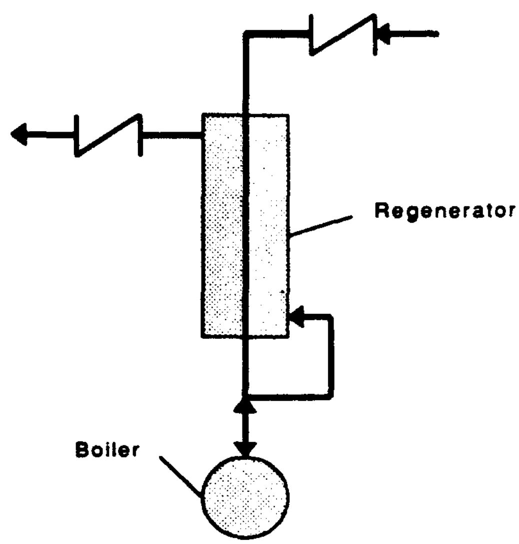
The additional degree of freedom presented by this change allows the boiler to be connected for "once-through" instead of reversing flow, as illustrated in Figure 10(c). The once-through arrangement was thought to be preferable to the dead-ended arrangement since it requires the boiler to be swept by a fresh charge of water in each cycle. This may help to prevent persistent dry-spots which could cause the boiler to overheat, and also flushes any non-condensible gases that might otherwise accumulate and interfere with condensation.

5.2.1 Analytical Model

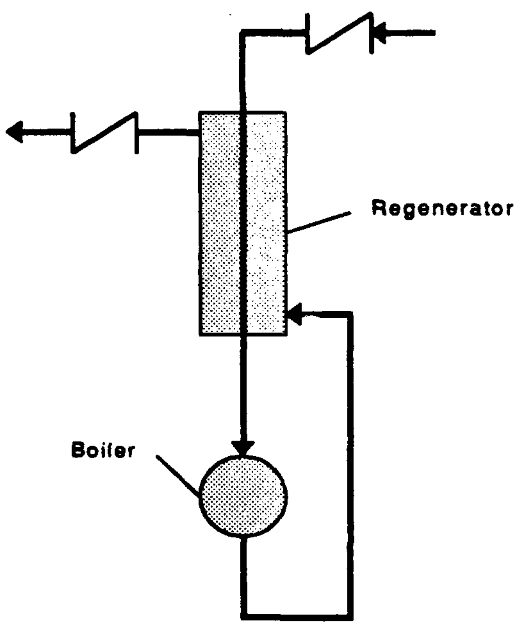
The change to a once-through boiler introduces a difficulty in modelling which was avoided by the original approach. While probably an oversimplification, the MK-I model assumes that a



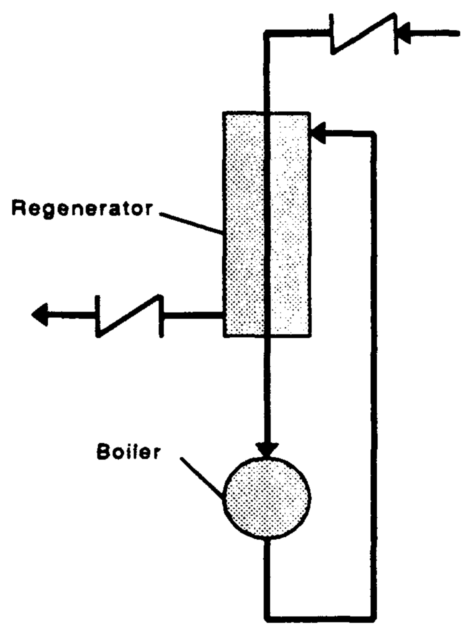
(a) Original Single-Tube Regenerator
with Dead-Ended Boiler



(b) Modified Tube-in-Tube Regenerator
with Dead-Ended Boiler



(c) Modified Tube-in-Tube Regenerator
with Once-Thru Boiler



(d) Same as Scheme (c) with
Parallel-Flow Regenerator

Figure 10. Alternate Flow Schemes

column of water enters the tubular boiler as a "liquid piston", compressing the steam ahead of it, and is then expelled by the compressed steam. Any sensible heat transfer to this water column is neglected in comparison to the latent heat transfer due to evaporation. In the new case of once-through boiler flow, a column of water is drawn into the boiler by the suction of the low-pressure steam contained in the boiler and regenerator outlet passage. After the incoming water column has been decelerated, the steam pressure must be increased by boiling in order to accelerate the outlet column and thus sustain the oscillations. However, to satisfy continuity, the steam must be displaced from the boiler by the incoming water column. We now have the problem of transporting the fresh liquid from the boiler inlet to the outlet. For a tubular boiler, the liquid will be transported from the inlet to the outlet by flashing of steam. The modelling of this process is quite complex, involving transient heat transfer and two-phase fluid dynamics. This problem was avoided in the original scheme, which simply assumed that the water reversed its flow direction in the boiler without considering any heat transfer to the water.

The MK-II model assumes that all liquid that enters the boiler is instantaneously transported to the outlet liquid column. Conceptually, this is as though a two-phase sub-cooled slug flow regime existed in which the liquid slugs traversed the 10 - 20 feet between the boiler inlet and its outlet adiabatically. While this is admittedly unrealistic, as a limiting case it was considered closer to reality than the alternative of assuming that all the water entering the boiler is evaporated and recondensed in the regenerator.

The MK-II model was run for a number of cases representative of the test apparatus with the new tube-in-tube heat exchanger. The model predicted that the modified apparatus should be capable of sustained oscillations and vigorous pumping. Examples of predicted pressure, displacement, and velocities are shown in Figures 11 and 12.

5.2.2 Experimental Results

In an effort to increase the outside HTC and thus the overall HTC in the regenerator, and at the same time provide the additional "thermal inertia" that was lacking in the original scheme, the regenerator was redesigned as a tube-in-tube heat exchanger. The redesigned regenerator is 20 feet long, and consists of a 1-1/8" OD copper tube with a concentric 3/4" OD inner tube. Depending upon the flow rate, the time-averaged convective HTC due to the oscillating water flow should be between 1000 - 2000 Btu/hr-ft^2-$^\circ$F. This should result in an overall HTC on the order of 1000 Btu/hr-ft^2-$^\circ$F.

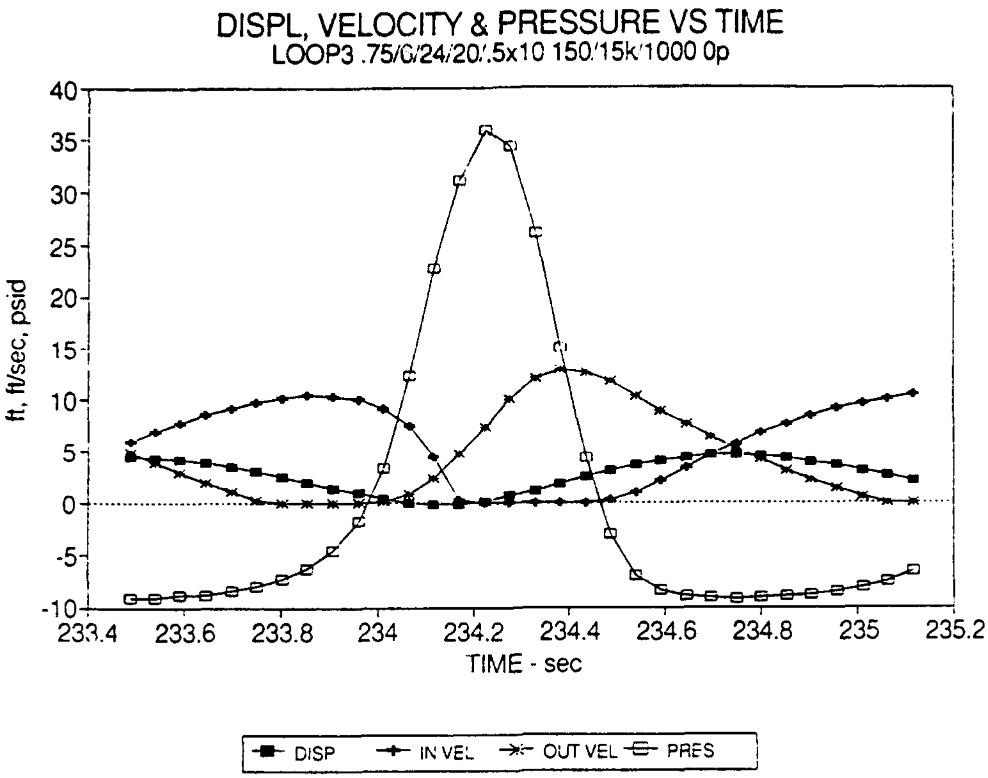


Figure 11. MK-II Cycle

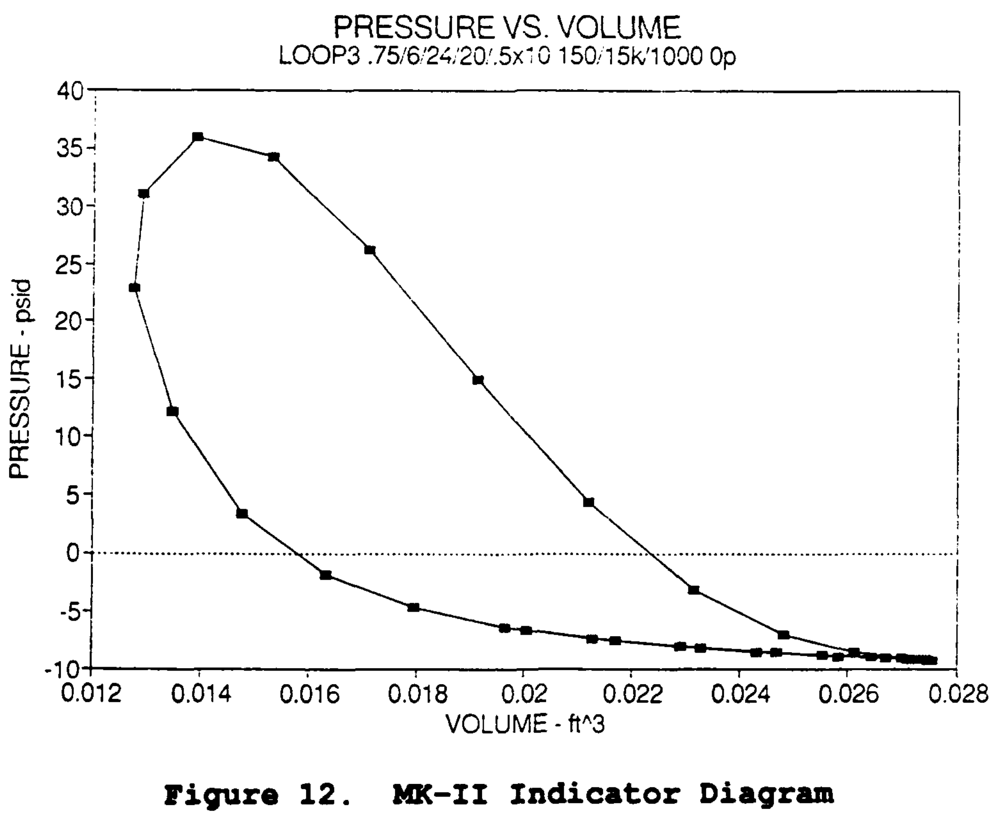


Figure 12. MK-II Indicator Diagram

The tube-in-tube regenerator was installed in the self-powered boiler test rig. All piping was 3/4" copper tube (7/8" OD), except for the inlet and outlet tubes, which retained the original 3/4" OD, 0.705" ID refrigeration tubing. The regenerator was connected to the boiler as shown in Figure 13. This produced a parallel-flow coolant path with the inlet to the boiler at the bottom, and the outlet at the top. A compound Bourdon gauge and a thermocouple were installed at the boiler outlet. Boiler temperature was measured by a thermocouple brazed to the boiler coil. Check valves were located at the inlet and at the outlet.

Initially when the burner was fired, the water displaced from the boiler flowed from the outlet. Subsequently bumping was observed, but there was little flow from the outlet and apparently no flow into the inlet. The boiler and boiler outlet remained at 215 - 225°F for a considerable length of time with little flow coming from the outlet. The burner was shut down when the temperature reached 500°F, and restarted when it dropped below 200 °F. Eventually, steam was produced at the outlet of the heat exchanger.

Next, approximately 24 ft of 3/4" OD refrigeration tubing was added to the outlet. This had no apparent effect.

Next, a ball valve was added at the outlet ahead of the check valve. The boiler pressure was allowed to build, and then the valve was opened suddenly. This resulted in some oscillatory flow from the outlet, but little inlet flow. The largest inlet flow occurred after the burner was shut down, due to condensation of the steam in the system.

Due to the fact that sustained flow oscillations could not be established, it was not possible to maintain cooling of the regenerator. Therefore, it was decided to cool the regenerator with an external supply of cooling water. It was hoped that this would isolate the hoped-for pumping effect from any deficiency in cooling.

Also, the inlet to the boiler was reconfigured with a bleed supply of water from the inlet tube (without passing through the regenerator), and with a check valve to prevent backflow. The main supply to the pump was teed into the junction of the regenerator and the outlet tube, as in the MK-I design. Thus, most of the water would enter the pump at the regenerator-outlet tube tee, while a smaller amount of water would be supplied as feedwater to the base of the boiler to prevent dryout of the boiler. These modifications are shown in Figure 14.

With this configuration, any pressure oscillations would cause water to be fed to the base of the boiler, thus hopefully preventing the dryout that forced previous tests to be

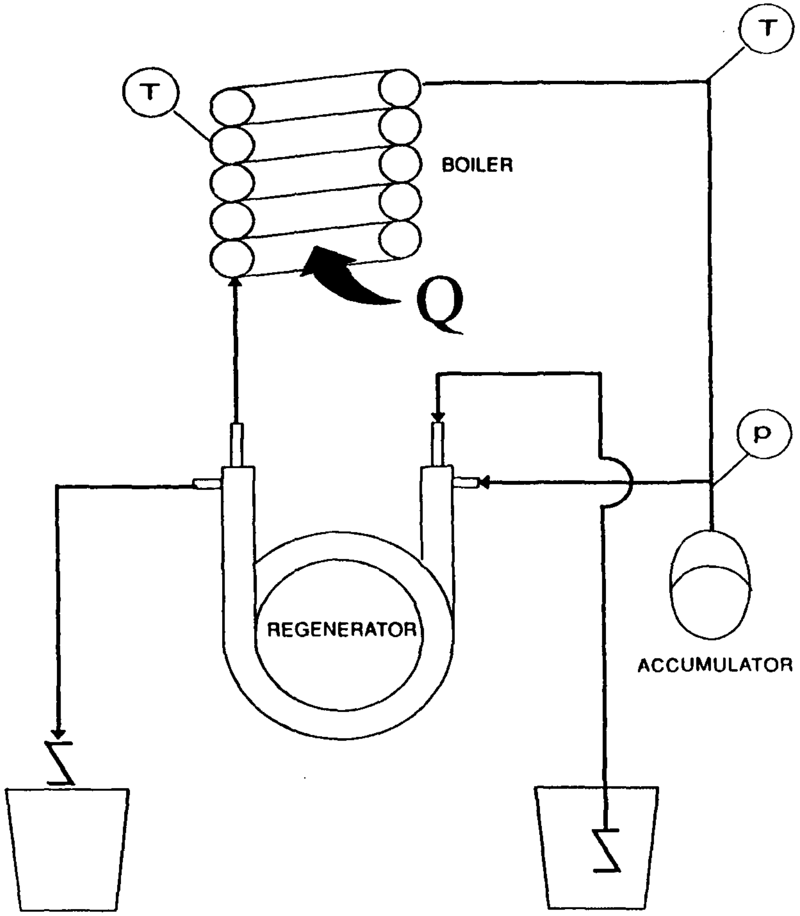


Figure 13. MK-II Initial Setup

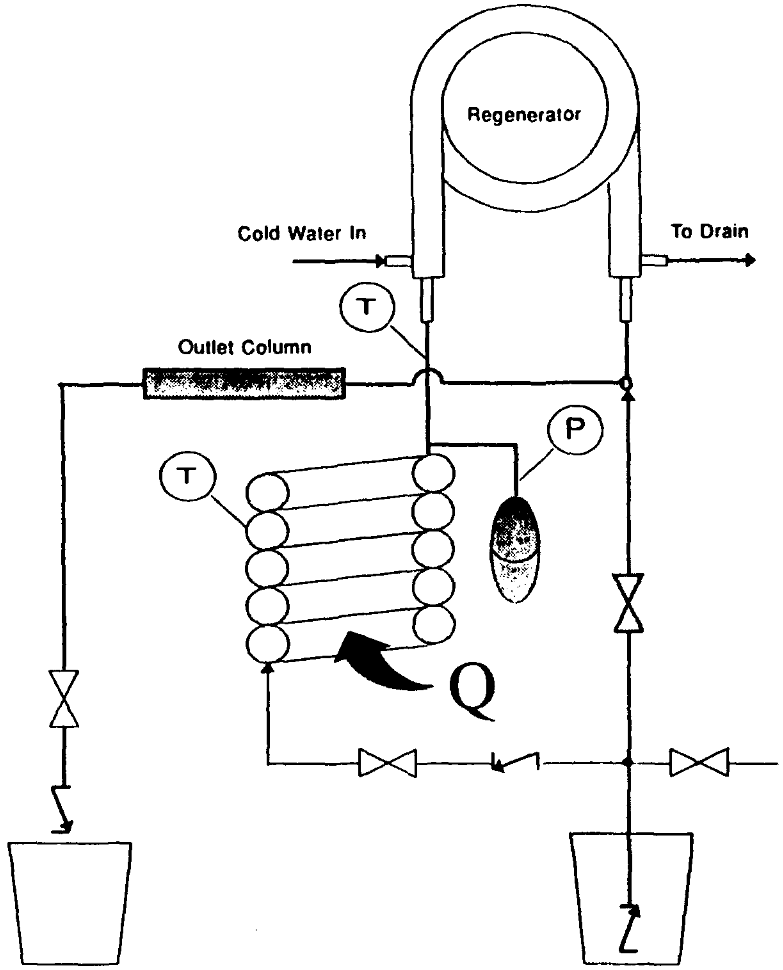


Figure 14. Modified MK-II Test Setup

terminated. In fact, with this change the boiler could be operated for approximately two minutes before it would overheat. During this period, some intermittent oscillations would be observed, but they would soon stop and the boiler would then overheat.

5.3 MK-III Pump

The discrepancy between the vigorous pumping predicted by the MK-I and MK-II analytical models versus the inability of the experimental apparatus to achieve sustained pumping led to a re-examination of the modelling assumptions. The MK-III analytical model was basically a re-formulation of the MK-I single-ended boiler analytical model, but with the single-tube periodic flow regenerator of the MK-I pump replaced by the tube-in-tube regenerator of the MK-II pump.

5.3.1 Analytical Model

A key assumption of the MK-I and MK-II models is that the steam phase is dry. No account is taken of the change in mass and energy of the liquid in the boiler. This analytical approach was taken in part because the actual two-phase dynamics and thermodynamics are in fact very complex, and in part because it was reasoned that the thermodynamics of the liquid phase could be neglected in comparison to the thermodynamics of the vapor. Conceptually, it was reasoned that the liquid phase thermodynamics could be lumped with the thermodynamics of the liquid piston.

An alternate assumption is to assume that the steam is a homogeneous equilibrium two-phase mixture of saturated liquid and vapor, i.e., wet steam. The MK-III model makes this assumption, and is otherwise identical to the MK-I model of a single-ended boiler, as illustrated in Figure 15. The wet steam is treated as a closed system of constant mass, expanding and contracting against the liquid piston while receiving heat at a constant rate from the boiler surface and rejecting heat at a variable rate to the wall of the regenerator. While it is recognized that the steam is not in fact a closed constant-mass system, this assumption is justified on the basis that in the periodic steady-state the average state of the steam is unchanged, and that such a model will properly track the thermodynamics of the liquid and vapor if the assumption of liquid-vapor equilibrium is to be invoked.

This modelling assumption introduces an additional variable that was avoided by the earlier models, that of the steam quality, $\mathbf{X}$. The value of $\mathbf{X}$ depends on the assumption regarding the total mass of steam in the boiler; in other words, how much liquid to include in the homogeneous liquid-vapor phase. As it will be seen, the detailed results are sensitive to the assumed value of **X**. However, the overall conclusion regarding the magnitude of the pumping effect is not sensitive to the assumed valve of **X**, but merely to the assumption regarding wet steam.

The energy equation for the steam is

$$m \times du = dQ - p \times dV \tag{1}$$

where:
- m = mass of steam = constant
- u = specific internal energy
- Q = net heat transfer
- p = steam pressure
- V = steam volume

Here, the Q is the <u>net</u> heat transfer, which is the heat received in the boiler less the heat removed by condensation in the regenerator, and V is the total steam volume, which is the volume of the boiler plus the volume of steam in the regenerator. This latter volume is dependent on the position of the liquid-vapor interface in the regenerator, or in other words, the displacement of the liquid piston.

If we are considering the steam to be dry, at low pressures we may write

$$\frac{du}{dT} \approx c_V \tag{2}$$

where:
- T = steam temperature
- c_V = constant-volume specific heat of steam $\approx$ 0.28 Btu/lb-°F

However, if we are dealing with two-phase steam,

$$du = du_f + X \times du_{fg} + u_{fg} \times dX \tag{3}$$

where:
- u_f = internal energy of liquid = $c_f \times dT$
- u_{fg} = internal energy of vaporization
- **X** = steam quality

If we consider the case of constant volume (dv = 0), then for steam near atmospheric pressure,

$$du \approx C_f \times dT + X \times T \times v_{fg} \times \frac{d^2p}{dT^2} \times dT \tag{4}$$

where:
- c_f = 1 Btu/lb-°F.
- T $\approx$ 672 °R
- $v_{fg} \approx$ 27 ft^3/lb
- $d^2p/dT^2 \approx$ 0.00087 Btu/ft^3-$°F^2$

In this case $du/dT \approx 1 + 15.8X$. Thus, the effective constant-volume heat capacity of wet steam is 3.5 to 60 times greater than dry steam. Consequently, the previous dry steam assumption substantially overestimates the temperature change (and resultant pressure change) due to heat transfer.

The effect of this change in the formulation of the model is to substantially reduce the thermal-to-mechanical energy conversion. A typical cycle and indicator diagram are shown in Figures 15 and 16. Both the pressure and displacement amplitudes are too low to be of practical interest. Oscillation frequency is 0.4 Hz. Analyses to determine the influence of the lengths and volumes of the boiler, regenerator, and inlet/outlet tubing indicate no significant improvement. While reducing the steam quality increases the work per cycle, resulting pressure and flow amplitudes are still too low to be of practical interest.

5.3.2 Experimental Results

In spite of the adverse prediction of the MK-III model, it was decided to proceed with testing of the model in order to either verify its validity or to determine how it could be modified.

Following the previous series of tests, it was observed that, in accordance with the analytical model, the boiler represents a source of steam and a compressible volume. In the event that the problem in obtaining sustained oscillations might be due to the lack of a uniform supply of steam, it was decided to operate the boiler with a controllable, independent supply of feedwater, and to supply steam to the pumping chamber in a controlled fashion. In this manner, any problems of steam supply could be isolated from the intended pumping effect.

The boiler was supplied from the city water supply (80 psig) through a throttle valve. A relief valve was used to regulate the boiler pressure to a maximum of 30 psig. A valve at the boiler outlet was used to throttle the steam to the pumping chamber. The required compressible volume was provided by a 2-inch ID x 12-inch long copper tube teed to the outlet of the boiler. The air could be vented from this chamber by a bleed valve at one end. The boiler outlet and steam chamber were teed to one end of the regenerator as shown in Figure 17. Thus, except for the fact that the steam was now supplied from an external source, the apparatus corresponded to the configuration of the MK-III analytical model.

The entire system was filled with water, and the boiler feed was set at approximately 0.05 - 0.10 GPM. The burner was fired at a rate of 30,000 - 35,000 Btu/hr. The stop valve at the exit

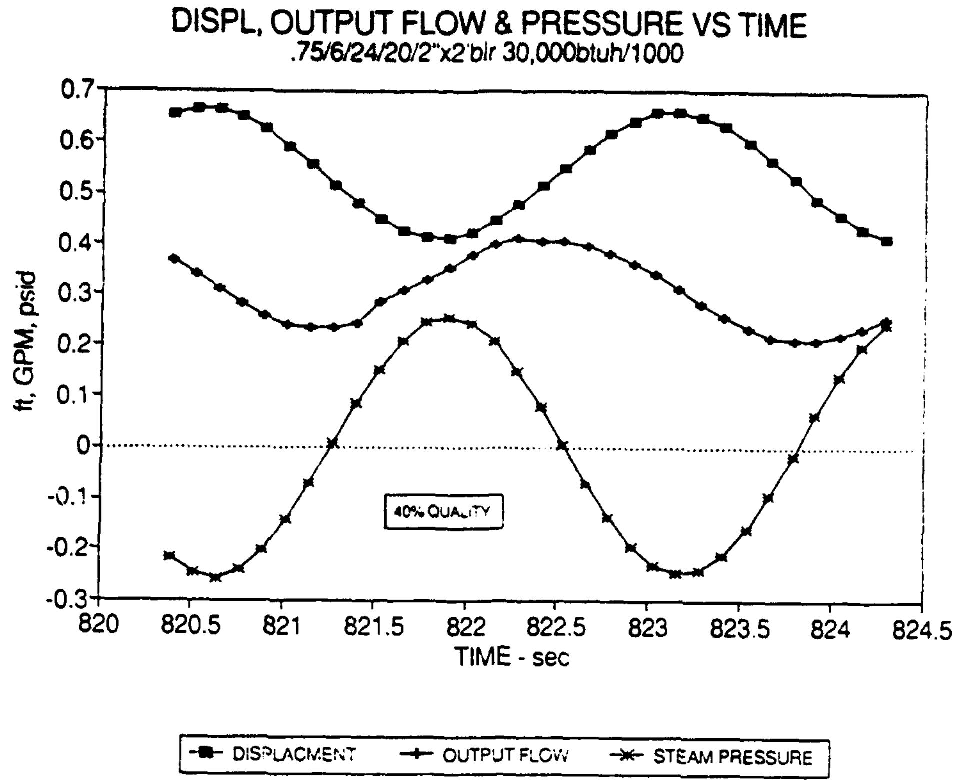


Figure 15. MK-III Cycle at 40% Quality

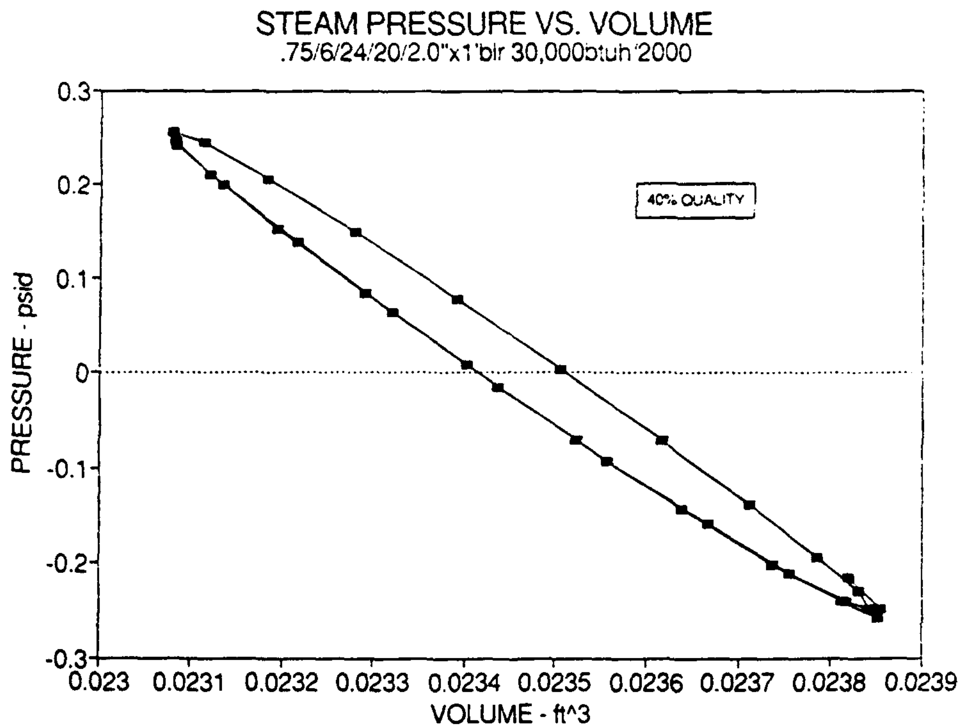


Figure 16. MK-III Indicator Diagram at 40% Quality

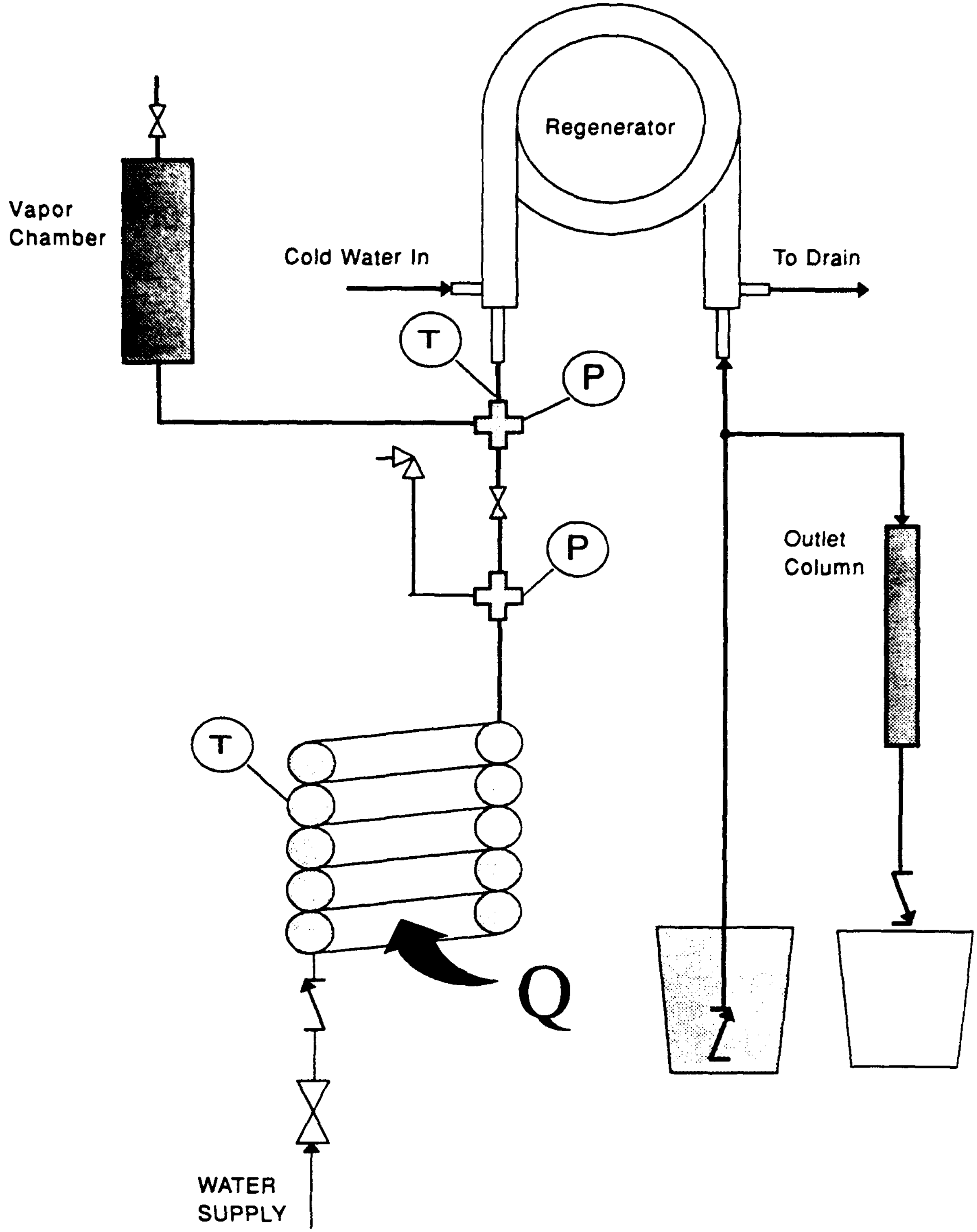


Figure 17. MK-III Test Setup with Externally-Supplied Steam Generator

of the outlet tube was closed and the system pressure was allowed to build to 30 psig. As the water drained from the steam chamber, the pressure oscillated violently, accompanied by banging sounds. After the banging subsided, the stop valve was opened. There would be a sizable flow of water from the outlet and the pressure would drop suddenly to below atmospheric pressure, accompanied by a smaller flow into the inlet tube. In most attempts, low amplitude sustained pressure and flow oscillations would begin immediately. However, the amplitude of these oscillations did not appear to be sufficient to open the inlet check valve. The outlet flow rate was measured by collecting the flow into a graduate, since it was too small to be measured with any accuracy by the weigh tank. The flow ranged from 0.05 - 0.2 GPM; therefore most or all of the outlet flow appeared to be supplied by condensation of the steam supplied by the boiler.

The results were not highly reproducible. Sometimes, after firing the boiler, the system would not oscillate and the system would have to be restarted. On other occasions, the cooling water would be manually adjusted to a lower flow rate, which would cause the amplitude of oscillation to diminish. When the cooling water flow would be returned to its original setting, the pressure amplitude would not increase to its original level. This lack of reproducibility causes the observations reported below to be somewhat subjective, in that they refer to trends rather than conclusively repeatable effects.

The degree of steam throttling was adjusted to vary the amount of steam being vented by the relief valve, and thus the amount of steam supplied to the pumping chamber. It appeared that a slight increase in amplitude was achieved when the steam flow was reduced from its maximum (i.e., at zero steam venting), after which the amplitude diminished as the steam flow to the pumping chamber was reduced to zero.

The flow of cooling water was varied from 1.6 GPM up to 6 GPM. The amplitude increased slightly at the high cooling flow.

Attempts were made to superheat the steam from the boiler. As it was difficult to regulate the flows of feedwater and steam, the amount of superheat could not be controlled. However, during one test, the steam temperature measured at the boiler outlet registered 300°F (i.e., approximately 88°F of superheat). This had no apparent effect on the oscillations.

A check valve was added to the exit of the outlet tube. This caused the oscillations to cease. Apparently, the pressure/flow amplitude was insufficient to open the check valve.

The length of the 2"-diameter steam chamber was increased from 12 inches to 2.5 feet. This did not improve the oscillations; if anything, the effect was negative.

A typical test result is shown in Figure 18. At approximately 320 seconds, the coolant flow is suddenly increased to 6 GPM, apparently causing some cooler water to reach the location of the thermocouple at the boiler outlet. At about 400 seconds, the steam begins to superheat, and at 470 seconds, the burner is shut off. The pressure amplitude is about +/- 2 psi, and the temperature amplitude is about +/- 2.5°F. At atmospheric pressure, dT/dp should be about 2.5°F/psi. Therefore, the measured temperature amplitude is only about half what would be predicted for the observed pressure amplitude. This apparent attenuation, as well as the phase lag of the temperature, is due to the transient heat conduction through the tube wall.

The measured frequency is about 0.42 Hz, compared to a predicted frequency of 0.4 Hz at 40% quality and 0.25 Hz at 5% quality. The measured pressure amplitude of +/- 2 psi compares to a predicted 0.25 psi at 40% quality (Figures 15 and 16), and 2 psi at 5% quality (Figures 19 and 20). Thus the observed frequency suggests that the high quality assumption applies, while the observed amplitude agrees more with the low quality assumption.

5.4 MK-IV Pump

In the original approaches (MK-I - MK-III), the liquid piston comprised both the output water column and the steam compression piston. It was observed that the steam volume and hence the pressure amplitude depended on the output liquid piston displacement. If the output (i.e., the flow rate) was restricted, the pressure amplitude would also be reduced, which would further reduce the output, and so on. It was reasoned that perhaps if the output piston were decoupled from the steam compression piston, a greater pressure amplitude might be developed independent of the output amplitude.

In a "Fluidyne" engine or pump[1], a "displacer piston" oscillates freely while displacing gas between hot and cold spaces. As illustrated in Figure 21, when the displacer moves toward the cold chamber, it displaces gas into the hot space, thereby increasing its pressure. The pressure acts against a second output piston to produce work or pumping. Since the displacement is a constant-volume process producing/absorbing no net work, theoretically the amplitude of the displacer piston can be much larger than that of the output piston. By the same

[1] C.D. West, Liquid Piston Stirling Engines, Van Nostrand Reinhold Company, New York, 1983.

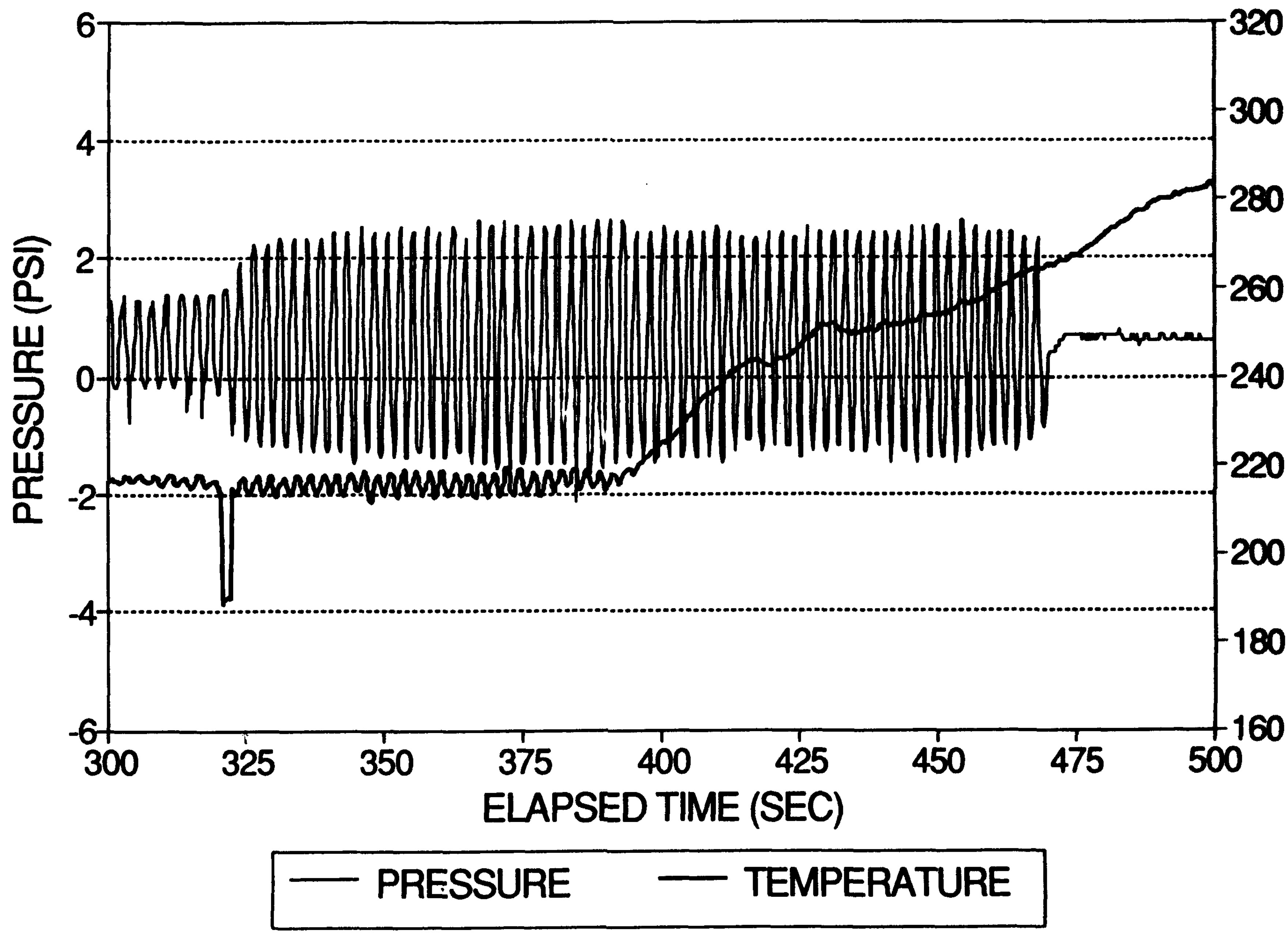


Figure 18. MK-III Test Results

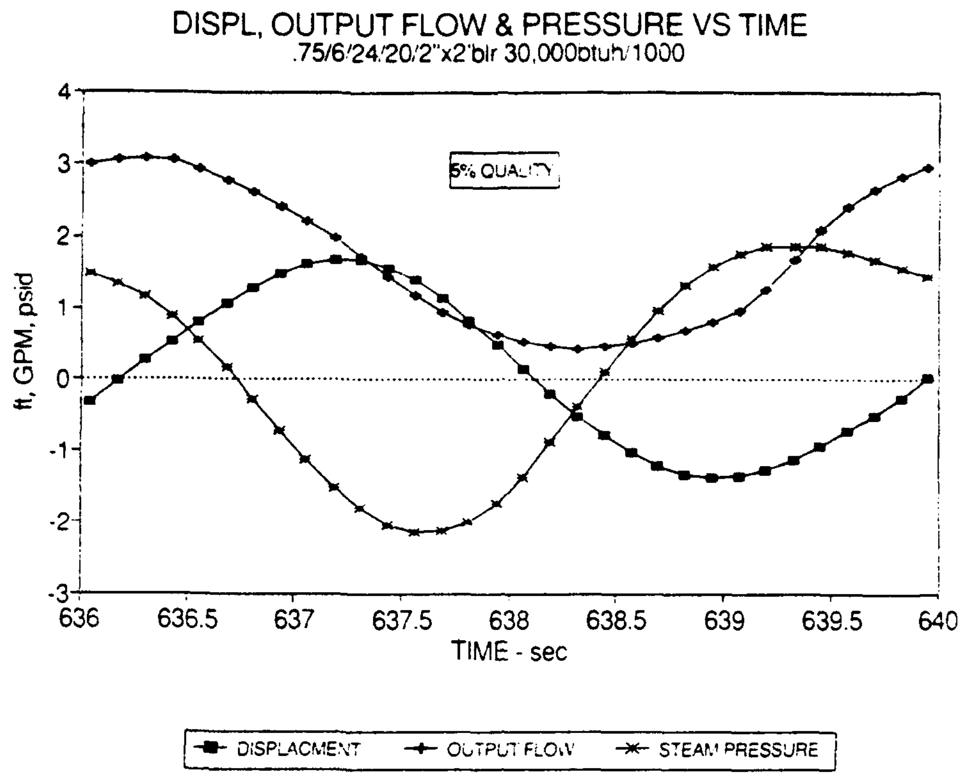


Figure 19. MK-III Cycle at 5% Quality

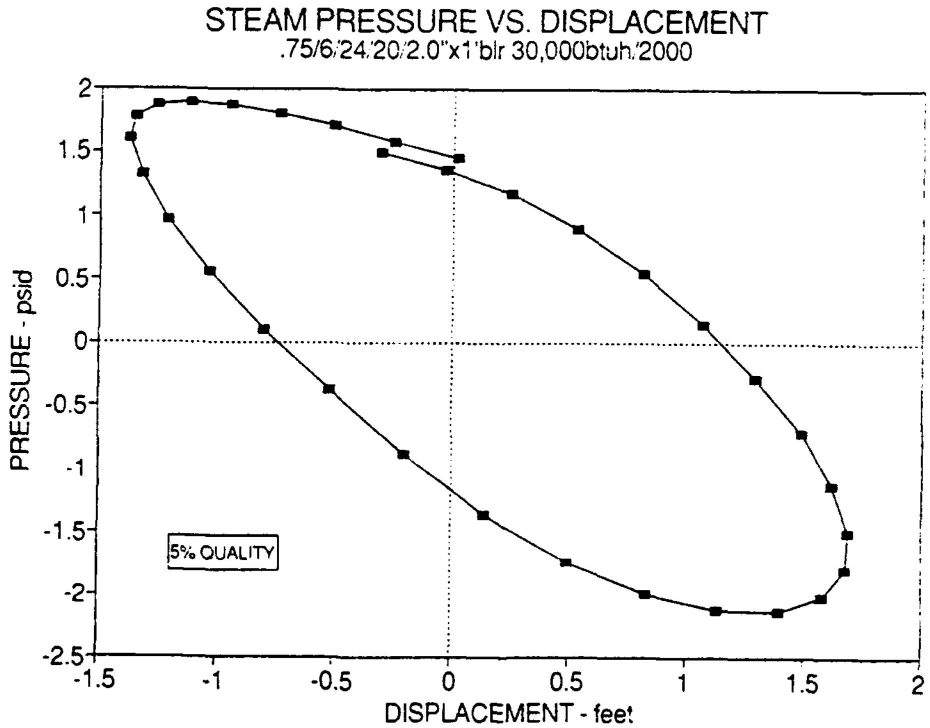


Figure 20. MK-III Indicator Diagram at 5% Quality

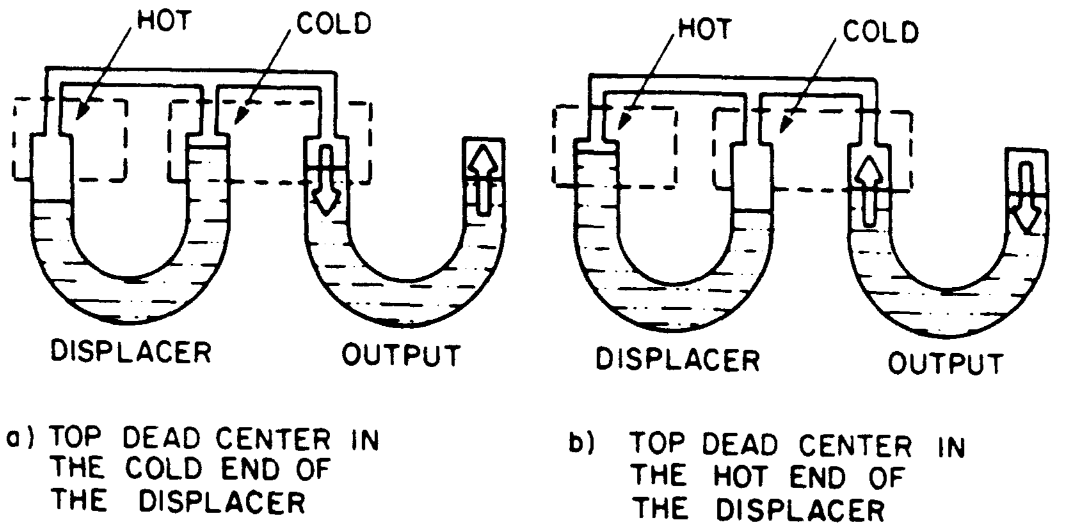


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Figure 21. Basic Operation of the Fluidyne (from Ref. 1)

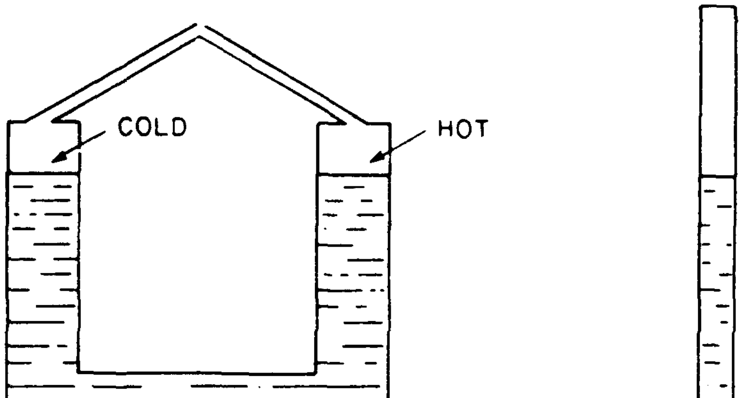

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Figure 22. Liquid Feedback Fluidyne (from Ref. 1)

token, some means must be employed to maintain the motion of the displacer piston, which would otherwise be slowed by the action of viscous friction. One way this is done is by allowing the output piston to transfer some of its kinetic energy to the displacer to maintain its motion, as illustrated in Figure 22.

Fluidyne engines and pumps all use a non-condensible working gas, although "wet" Fluidynes will also permit some evaporation/condensation of the liquid pistons. Dry Fluidynes have low specific power but are more efficient than wet Fluidynes, which are limited to about 1% thermal efficiency. Low efficiency is not a concern in a water heating application, however, as long as all the energy is used to heat the water. Since heat transfer coefficients are fairly low in convection to gases, Fluidyne engines and pumps require substantial heat transfer surface area. However, a Fluidyne pump that relied on boiling and condensation of water could take advantage of large heat transfer coefficients, and could therefore require relatively little heat transfer area.

The MK-IV pump based on these considerations is shown in Figure 23. The displacer piston is contained within the "U"-tube on the left. One leg of the U-tube is surrounded by a cooling water jacket. In practice, the cooling water would be the water being pumped/heated. In the laboratory apparatus, an external supply of cooling water is used.

A boiler supplies steam to the space above the displacer liquid piston. If located sufficiently below the liquid/vapor interface, the boiler feed can be supplied by gravity. In the laboratory apparatus, an external boiler feed is provided for. The steam from the boiler pressurizes both legs of the displacer liquid piston. If the displacer piston is set into oscillation toward the cold leg, it will cover more of the cold surface within the cooling jacket, thereby reducing the rate of condensation and causing the pressure to rise. On the reverse stroke, more of the condenser will be uncovered, causing the steam to condense and the pressure to drop.

The output tube is teed into the base of the hot leg. When the steam pressure is greater than atmospheric pressure (less the hydrostatic head), both legs of the displacer are forced toward the outlet, thus pressurizing the output tube.

5.4.1 Analytical Model

The analytical model for the MK-IV pump is shown in Appendix A. The steam thermodynamics follows the treatment of MK-III, that is, it assumes wet steam. The liquid piston dynamics are also similar to the previous models, that is, one-dimensional flow with viscous friction.

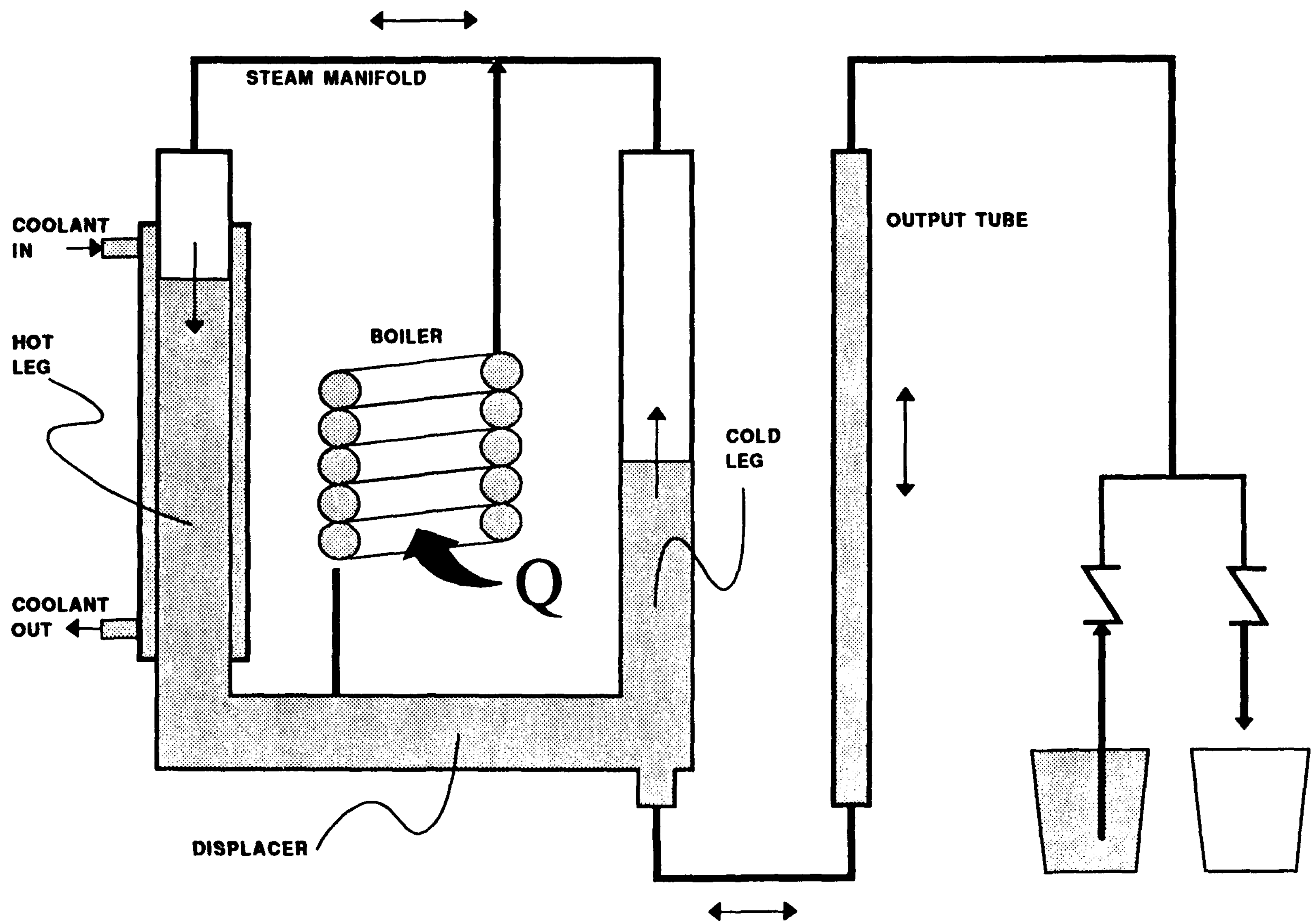

Figure 23. MK-IV Pump Model

A key assumption in the model is the modelling of the interaction between the liquid pistons comprising the displacer and the output column. This interaction depends on the detailed configuration of the tee. For preliminary analysis, the condition at the tee was simply assumed to be uniform pressure; that is, the static pressure of each of the hot leg, cold leg and output column is identical at the output tee. With this assumption, both the hot and cold legs move in phase with the output column; that is, they do not oscillate with the rocking motion of a U-tube. In order for the model to predict out-of-phase oscillations of the hot and cold legs of the displacer, it was necessary to introduce a forcing function that is out-of-phase with the pressure. This was done by assuming that the pressure acting at the base of the hot leg of the displacer during its up-stroke is equal to the stagnation pressure of the output piston, rather than its static pressure. There is no theoretical basis for this assumption, only that we wish to induce an out-of-phase motion. However, there is some justification in this assumption from the observation reported by West[1], that in order to induce oscillatory displacer motion in a Fluidyne engine, the output tube must face in the direction of the hot leg.

With the above "stagnation-pressure" model, a small phase difference is introduced between the motion of the hot and cold legs of the displacer. Nonetheless, there is negligible difference between the pumping action with and without the phase lag. A typical result is shown in Figures 24 and 25 for the case of a 12′ x 0.75" ID output piston. The steam pressure amplitude is about +/- 0.2 psi, and the flow amplitude is about +/- 0.5 GPM (This is without check valves, i.e., pumping against zero head). The frequency is about 0.5 Hz. The indicator diagram in Figure 25 shows that the expansion work is about the same as the compression work, which is a consequence of the small phase difference between the pressure and steam volume (the displacement of the output piston is almost 180° out of phase with the steam pressure. Increasing the length of the output piston to 24 ft (to reduce its natural frequency) causes the oscillation to cease. Reducing its length to 6 ft causes the displacer to oscillate as a U-tube, that is, with the desired 180° phase difference between the motion of the hot and cold legs. However, the amplitudes are extremely small.

The analysis, therefore, gives the surprising result that the intended rocking displacer motion would result in negligibly small pumping amplitudes, while suppression of its rocking motion provides the largest, although disappointingly small, amplitudes. This is quite surprising in view of the fact that Fluidyne water pumps of this type have been successfully developed. We can only surmise that this conclusion would be different if we had considered the presence of a non-condensible working gas in addition to the steam, which is the case with successful Fluidyne

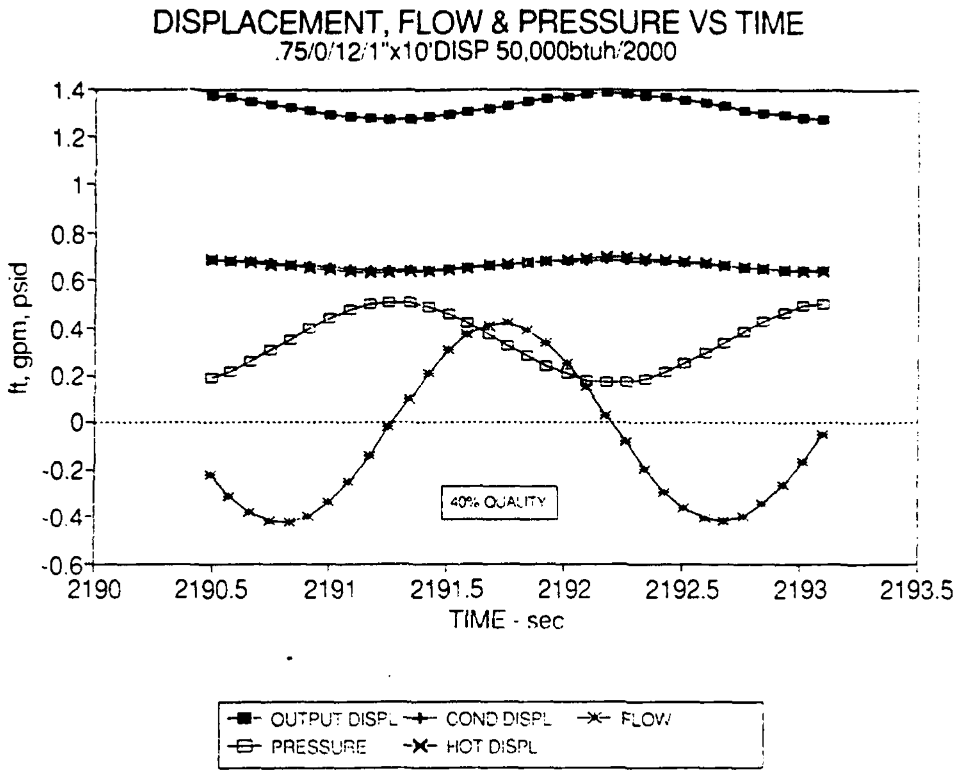

Figure 24. MK-IV Cycle

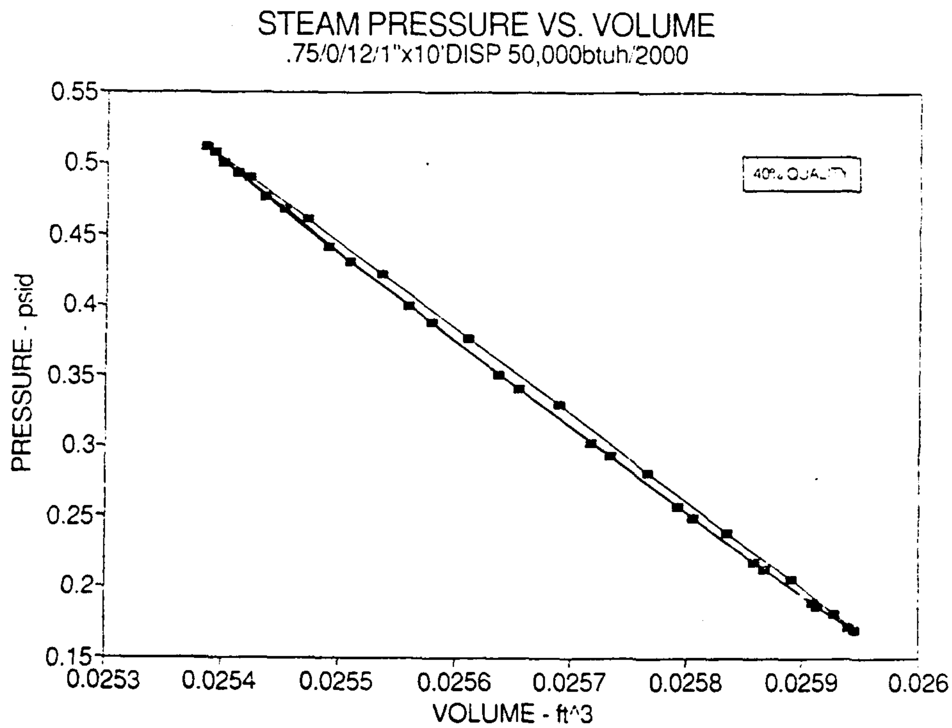

Figure 25. MK-IV Indicator Diagram

pumps. However, this non-condensible gas would preclude the boiler approach and would require a substantially larger heater/cooler surface area.

5.4.2 Experimental Results

Despite the negative predictions of the MK-IV analytical model, it was decided to test the MK-IV configuration on the chance that conclusions of the analysis were wrong. The apparatus was constructed of copper tubing and copper fittings, as illustrated in Figure 26. The stainless steel boiler that had been used in the previous apparatus was used as the steam supply. The boiler could be gravity-fed from the displacer piston, or could be fed from an external supply. Since the gravity feed did not produce a steady rate of feed, the external water supply was used in the tests. In order to provide the best opportunity for oscillations to amplify, the inlet and outlet check valves were removed, and the mouth of the output tube was immersed in a flooded graduate. The dimensions of the key elements of the apparatus are listed in Table 2 below.

Table 2. MK-IV Specifications

Displacer Tube:	
Hot/Cold Leg Height	54 in
Horizontal Leg Length	12 in
I.D.	0.995 in
O.D.	1.125 in
Cooling Jacket:	
Length	46 in
I.D.	1.505 in
Steam Manifold:	
Length	12 in
I.D.	0.811 in
Output Tube:	
Fixed Length	54 in
I.D.	0.811 in
Additional Length	7.5 - 24 ft
I.D.	0.375 - 0.680 in
Boiler:	
Heated Length	9 ft
I.D.	0.495 in
Firing Rate	34 - 60 MBTUH

The initial test was run at a firing rate of 34,000 BTUH and an additional 0.680" output tube length of 24 ft. The cooling jacket was supplied by 8 GPM of city water. The boiler was supplied with city water at approximately 0.05 - 0.1 GPM. Initially, the apparatus was completely filled with water and the boiler was fired. After steam was produced by the boiler, water would be displaced from the apparatus. Subsequently, a pulsating

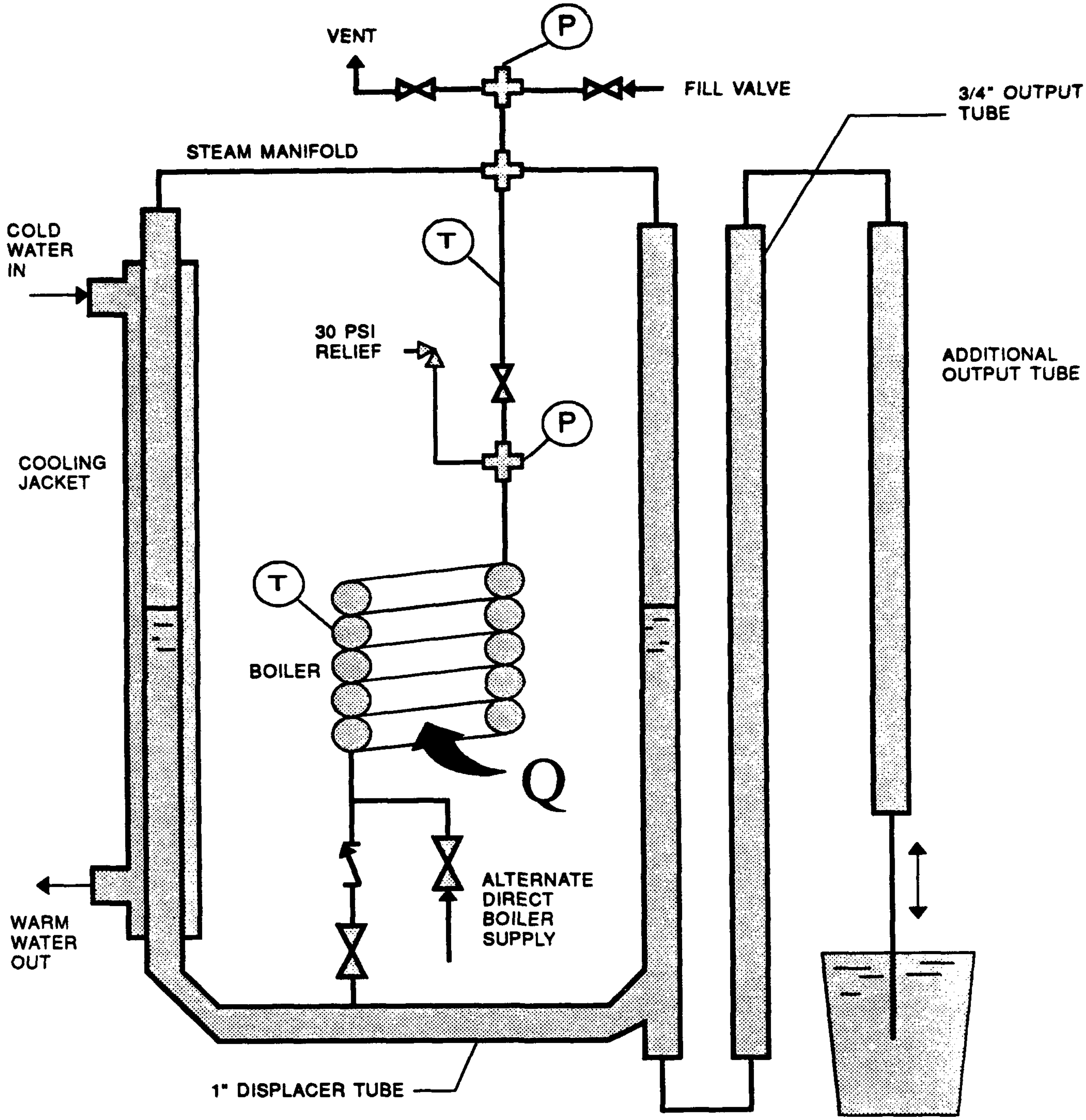

Figure 26. MK-IV Pump

flow of water would issue from the mouth of the output tube at an average rate equal to the boiler feed rate. The frequency of the pulsating flow was variable, but was in the range of 0.3 - 0.5 Hz. The amplitude of the displacement was in the range of 1 - 10 cu-in. The pressure amplitude was less than 0.1 psi, below the resolution of the digital data recorder.

Additional tests were run at higher and lower cooling water flow rates, a shorter 6.5 ft x 0.375 in (additional) output tube, and at a firing rate of 60,000 BTUH. There was no significant difference in the results.

6.0 CONCLUSIONS & RECOMMENDATIONS

1. While the MK-I and MK-II analytical models predict that the thermally activated pumping approach could produce a practical pumping effect, these predictions do not agree with subsequent experimental results. It is believed that the basic flaw in the formulation of these models is the assumption that the steam remains dry, i.e., 100% quality.

2. The MK-III wet steam model gives reasonable agreement with the experimental results. This model and the experimental results indicate that the proposed dynamic interaction between steam and a liquid piston is not capable of producing the desired pressure amplitude and flow in a practical device.

3. The MK-IV modification of introducing an oscillating displacer piston into the device did not improve the pumping performance. Rather, it appeared that the displacer motion diminishes pumping performance.

4. Based upon the conclusions of this research, it does not appear that further research and development of thermally activated water pumps based upon dynamic interaction of steam and one or more liquid pistons is warranted. Other heat-actuated pumps that are based upon hydrostatic principles rather than liquid-piston dynamics may be more successful, although more complex.

This document reports research undertaken at the U.S. Army Natick Research, Development and Engineering Center and has been assigned No. NATICK/TR-94/025 in the series of reports approved for publication.

APPENDIX Analytical Models of MK-I - MK-IV Pumps

Three-Port model of Thermally Activated Pump:
Simultaneous flow through inlet and outlet.
Zero Friction.

5	TIME STEP	sec		0.0035
6	TIME	SEC		120.3520
7	DESIGN PARAMETERS:			
8	INLET ID	in		1
9	INLET LENGTH	ft		6
10	OUTLET ID	in		1
11	OUTLET LENGTH	ft		18
12	INLET TEMPERATURE	deg F		60
13	DENSITY	#/ft^3		62.4
14	REGEN ID	in		1
15	REGEN LENGTH	ft		20
16	BOILER ID	in		1
17	BOILER LENGTH	ft		2.5
18	COLD WALL TEMPERATUR	deg F		70
19	HOT WALL TEMPERATURE	deg F		140
20	AVERAGE WALL TEMP	deg F		500
21				
22	OUTLET COLUMN:			
23	CROSS-SECTION	ft^2	@PI*H10^2/(4*144)	
24	VELOCITY	ft/sec	(H58*H56-H40*H35)/H23	
25	ACCELERATION	ft^2/sec	(H59*H56-H39*H35)/H23	
26	PRESSURE OUT (P6)	psig		0
27	PRESSURE IN (P5)	psig	(+H26+H32+H13*H11*H25/(32.2*144)+H27)/2	
28	STAGN. PRES (P5')	psig	+H27+H13*G24^2/(2*32.2*144)	
29	FLOW RATE	GPM	+H24*H23*60*7.48	
30	REYNOLDS NUMBER		(3600*H13*H24*H10/(12*0.74))*@ABS(H24)/H24	
31	FRICTION FACTOR			0
32	FRICTION LOSS		4*H31*(H11*12/H10)*H13*H24^2/(32.2*2*144)	
33				
34	REGENERATOR DYNAMICS:			
35	CROSS-SECTION	ft^2	@PI*H14^2/(4*144)	
36	FLOW DIRECTION		@IF(G40>0,"IN","OUT")	
37	FRICTION FACTOR			0
38	FRICTION LOSS	psid	4*H37*(H41*12/H14)*(H75^2/(2*H13*418000000*144))*(H40/@ABS(H40))	
39	ACCELERATION	ft^2/sec	144*32.2*(H51-H42-H38)/(H13*G41)	
40	VELOCITY	ft/sec	+G40+H39*H5	
41	INTERFACE LENGTH	ft	+G41+H40*H5	
42	STEAM PRESSURE (P4)	psig	+H103-14.696	
43	FLOW RATE	GPM	+H40*H35*60*7.48	
44				
45	MIXER:			
46	The three way mixing is assumed to occur in a tee having the same run diameter			
47	as the regenerator. The regenerator and outlet are connected to the run of the tee;			
48	the inlet is connnected via the branch.			
49				
50	OUTLET PRESSURE (P5)		+H27	
51	REGEN PRESSURE (P3)		+H50+H13*((G24*H23/H35)^2-G40^2)/(32.2*144)	
52	BRANCH PRESSURE (P1)		(@MAX(H50,H51)+4*H52)/5	
53	FLOW ERROR	GPM	+H43-H63+H29	
54				
55	INLET COLUMN:			
56	CROSS-SECTION	ft^2	@PI*H8^2/(4*144)	
57	ACCELERATION	ft^2/sec	(144*32.2*(H61-H62-H66)/(H13*H9)+H57)/2	
58	VELOCITY	ft/sec	@MAX(G58+H57*H5,0)	
59	ACCELERATION	ft^2/sec	@IF(H58>0,H57,0)	
60	STAGN. PRES IN (P0')	psig		0
61	PRESSURE IN (P0)	psig	+H60-H13*G58^2/(2*32.2*144)	
62	PRESSURE OUT (P1)	psig	(+H52+H62)/2	
63	FLOW RATE	GPM	+H58*H56*60*7.48	
64	REYNOLDS NUMBER		@MAX(3600*H13*H58*H8/(12*0.74),1)	

65	FRICTION FACTOR			0
66	FRICTION LOSS		@IF(H58>0,4*H65*(H9*12/H8)*H13*H58^2/(32.2*2*144),0)	
67	The water column in the regenerator is assumed to have no evaporation/condensation			
68	occurring at the liquid/vapor interface. The wall temperature and water temperature			
69	distributions are assumed to be linear, with uniform Q/A (i.e., uniform			
70	delta T).			
71	VISCOSITY	#/hr-ft		0.74
72	THERMAL CONDUCTIVITY	Btu/hr-ft-F		0.394
73	SPECIFIC HEAT	Btu/#-F		1
74	PRANDTL NUMBER		+H71*H73/H72	
75	MASS VELOCITY	#/ft^2-hr	@ABS(+H13*H40*3600)	
76	FLOW RATE	#/hr	+H75*H35	
77	REYNOLDS NUMBER		+H75*H14/(12*H71)	
78	HEAT TRANSFER COEF.	Btu/hr-ft^2-	@IF(H77>2300,0.023*(H72*12/H14)*H77^0.8*H74^0.4,4.36*H72*12/H14)	
79	WATER SURFACE AREA	ft^2	@PI*H14*H41/12	
80	WATER MASS	#	+H35*H41*H13	
81	NTU		+H78*H79/(H76*H73)	
82	WALL TEMP @ INTERFACE	deg F	+H18+(H19-H18)*H41/H15	
83	AVERAGE WALL TEMP	deg F	(H82+H18)/2	
84	COLD WATER TEMP	deg F	+H18-H86	
85	INTERFACE WATER TEMP	deg F	+H82-H86	
86	AVERAGE DELTA T	deg F	+H83-H92	
87	HEAT TRANSFER RATE	Btu/hr	+H78*H79*H86	
88	HEAT TRANSFERRED	Btu	+H87*H5/3600	
89	CUMUL HEAT TRANSFER	Btu	+G89+H88	
90	ENTHALPY FLUX	Btu	@IF(H36="IN",(H80-G80)*H73*H12,(H80-G80)*H73*H84)	
91	TOTAL ENERGY	Btu	+G91+H90+H88	
92	AVERAGE TEMPERATURE	deg F	+H91/(H73*H80)	
93	REGENERATOR THERMODYNAMICS -			
94	The steam column in the regenerator is			
95	assumed to be at a uniform pressure and at the same temperature as the steam in			
96	the boiler. Only the vapor phase is considered: The dynamics/thermodynamics of			
97	the condensed water is ignored. Thus, the steam phase is a open control volume			
98	whose mass is changing via flow in/out of the regenerator and in/out of the			
99	condensed water phase.			
100	INTERFACE VELOCITY	ft/sec	+H40	
101	BOILING LENGTH	ft	+H15-H41	
102	SURFACE AREA	ft^2	+H101*@PI*H14/12	
103	PRESSURE	psia	+H134	
104	STEAM TEMPERATURE	deg F	+H157	
105	AVERAGE WALL TEMP	deg F	(H19+H82)/2	
106	DENSITY	#/ft^3	+H156	
107	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H104+2.53E-08*H104^2	
108	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H104+0.006	
109	SPECIFIC HEAT	Btu/#-F	+H111*H108/H107	
110	LATENT HEAT	Btu/lbm	1062.3275+35.1711+(0.41242-1.023)*H104+0.00022*H104^2-9.517E-07*H104^3	
111	PRANDTL NUMBER			0.95
112	MASS VELOCITY AT INTER	#/hr-ft^2	@ABS(+H40*H106*3600)	
113	MASS VELOCITY AT EXIT	#/hr-ft^2	+H112*(H17)/(H15+H17-H41)	
114	AVERAGE MASS VELOCITY	#/hr-ft^2	(H112+H113)/2	
115	REYNOLDS NUMBER		+H114*H14/(12*H107)	
116	CONVECTIVE HTC	Btu/ft^2-hr-	@IF(H115>2300,0.023*(12*H108/H14)*H115^0.8,4.36*12*H108/H14)	
117	CONDENSING HTC	Btu/ft^2-hr-		2000
118	BOILING HTC	Btu/ft^2-hr-		2000
119	BOIL/COND/CONV?		@IF(H105>H104+10,"BOIL",@IF(H105>=H104,"CONV",@IF(H116*(H104-H105)>H117,"	
120	HTC	Btu/ft^2-hr-	+H116+H118*(1-@EXP(-(H121/10)^20))	
121	EFFECTIVE DELTA T	deg F	+H105-H104	
122	HEAT TRANSFER RATE	Btu/hr	+H120*H102*H121	
123	HEAT TRANSFERRED	Btu	+H122*H5/3600	
124	CUMUL HEAT TRANSFER	Btu	+H123+G124	
125	EVAPORATION	#	+H123/H110	
126	BOILER THERMODYNAMCIS:			
127	The boiler is assumed to be at a uniform pressure and temperature. As with the			
128	regenerator, only the vapor phase is considered. The mass of the vapor phase changes			

129	with flow in/out of the regenerator and through evaporation/condensation.			
130	The wall temperature is assumed to be uniform.			
131	CROSS SECTION	ft^2	@PI*H16^2/(4*144)	
132	INLET VELOCITY	ft/sec	(H100*H113/H112)*(H35/H131)	
133	INLET MASS VELOCITY	#/ft^2-hr	@ABS(H132)*H106*3600	
134	PRESSURE	psia	+H158	
135	SURFACE AREA	ft^2	@PI*H16*H17/12	
136	DENSITY	#/ft^3	+H156	
137	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H157+2.53E-08*H157	
138	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H157+0.006	
139	SPECIFIC HEAT	Btu/#-F	+H141*H138/H137	
140	LATENT HEAT	Btu/lbm	+H110	
141	PRANDTL NUMBER			0.95
142	AVERAGE MASS VELOCITY	#/hr-ft^2	+H133/2	
143	REYNOLDS NUMBER		+H142*H16/(12*H137)	
144	CONVECTIVE HTC	Btu/ft^2-hr-	@IF(H143>2300,0.023*(12*H138/H16)*H141^0.8,4.36*12*H138/H16)	
145	CONDENSING HTC	Btu/ft^2-hr-		2000
146	BOILING HTC	Btu/ft^2-hr-		2000
147	BOIL/COND/CONV?		@IF(H20>H157+10,"BOIL",@IF(H20>=H157,"CONV","COND"))	
148	HTC	Btu/ft^2-hr-	@IF(H147="COND",+H145,+H144+H146*(1-@EXP(-(H149/10)^20)))	
149	EFFECTIVE DELTA T	deg F	@MIN(+H20-H157,150)	
150	HEAT TRANSFER RATE	Btu/hr	+H148*H135*H149	
151	HEAT TRANSFERRED	Btu	+H150*H5/3600	
152	CUMUL HEAT TRANSFER	Btu	+H151+G152	
153	EVAPORATION	#	+H151/H140	
154	TOTAL MASS OF STEAM	#	+G154+H153+H125	
155	TOTAL VOLUME	ft^3	+H131*H17+H35*H101	
156	STEAM DENSITY	#/ft^3	(@MAX(+H154/H155,0.0001)+H156)/2	
157	SATURATION TEMPERATU	deg F	453.8761+88.4691*@LN(H156)+4.6261*@LN(H156)^2	
158	SATURATION PRESSURE		(0.496-0.023*H157+0.0004241*H157^2-2.667E-06*H157^3+1.09E-08*H157^4+7.767E	
159	OVERALL HEAT TRANSFER	Btu	+H152+H124+H89	
160	HEAT TRANSFER RATE	Btu/hr	3600*H159/H6	

2	TIME	SEC		235.1710
3	TIME STEP	sec		0.0025
4	DESIGN PARAMETERS:			
5	INLET ID	in		0.75
6	INLET LENGTH	ft		6
7	OUTLET ID	in		0.75
8	OUTLET LENGTH	ft		24
9	INLET TEMPERATURE	deg F		60
10	DENSITY	#/ft^3		62.4
11	REGEN ID	in		0.75
12	REGEN LENGTH	ft		20
13	BOILER ID	in		0.5
14	BOILER LENGTH	ft		10
15	COLD WALL TEMPERATUR	deg F		100
16	HOT WALL TEMPERATURE	deg F		200
17	HEAT TRANSFER RATE	Btu/hr		15000
18	OUTLET PRESSURE	psig		
19				
20	INLET DYNAMICS:			
21	CROSS-SECTION	ft^2	@PI*H11^2/(4*144)	
22	FRICTION FACTOR		0.046/H57^0.2	
23	FRICTION LOSS	psid	4*H22*((H6+H12)*12/H11)*(H55^2/(2*H10*418000000*144))	
24	ACCELERATION	ft^2/sec	144*32.2*(H29-H27-H23)/(H10*(H6+H12))	
25	VELOCITY	ft/sec	@MAX(+G25+H24*H3,0)	
26	ACCELERATION	ft^2/sec	@IF(H25>0,H24,0)	
27	STEAM PRESSURE (P4)	psig	+H68-14.696	
28	STAGN. PRES IN (P0')	psig		
29	PRESSURE IN (P0)	psig	+H28-H10*G25^2/(2*32.2*144)	
30	FLOW RATE	GPM	+H25*H21*60*7.48	
31				
32	OUTLET DYNAMICS:			
33	Water entering the boiler is assumed to be transported instantaneously to the			
34	outlet column of the regenerator, thus adding to interface length.			
35	CROSS-SECTION	ft^2	@PI*H7^2/(4*144)	
36	FRICTION FACTOR		0.046/H38^0.2	
37	FRICTION LOSS		4*H36*((G42+H8)*12/H7)*H10*H40^2/(32.2*2*144)	
38	REYNOLDS NUMBER		@MAX((3600*H10*H40*H7/(12*0.74)),1)	
39	ACCELERATION	ft^2/sec	144*32.2*(H43-H37-H44)/(H10*(H8+G42))	
40	VELOCITY	ft/sec	@MAX(+G40+H39*H3,0)	
41	ACCELERATION	ft^2/sec	@IF(H40>0,H39,0)	
42	INTERFACE LENGTH	ft	+G42-H40*H3+H25*H3	
43	STEAM PRESSURE (P4)	psig	+H27	
44	PRESSURE OUT (P6)	psig	+H18	
45	FLOW RATE	GPM	+H40*H35*60*7.48	
46				
47	INLET WATER THERMO:			
48	The water thermodynamics are not modelled explicitly. Inlet water is assumed to			
49	be heated in the regenerator, and then partially flashed in the boiler. The inlet			
50	water is then displaced to the outlet column.			
51	VISCOSITY	#/hr-ft		0.74
52	THERMAL CONDUCTIVITY	Btu/hr-ft-F		0.394
53	SPECIFIC HEAT	Btu/#-F		1
54	PRANDTL NUMBER		+H51*H53/H52	
55	MASS VELOCITY	#/ft^2-hr	@ABS(+H10*H25*3600)	
56	FLOW RATE	#/hr	+H55*H21	
57	REYNOLDS NUMBER		@MAX(+H55*H11/(12*H51),1)	
58	HEAT TRANSFER COEF.	Btu/hr-ft^2-	@IF(H57>2300,0.023*(H52*12/H11)*H57^0.8*H54^0.4,4.36*H52*12/H11)	
59	WATER SURFACE AREA	ft^2	@PI*H11*H12/12	
60	NTU		+H58*H59/(H56*H53)	
61	OUTLET STEAM THERMO:			
62	The steam column in the regenerator is assumed to be at a uniform pressure			
63	and at the same temperature as the steam in the boiler. The average wall temperature			
64	for condensation in the regenerator is assumed to be constant.			

65	OUTLET INTERFACE VELO	ft/sec	+H40	
66	STEAM LENGTH	ft	+H12-H42	
67	SURFACE AREA	ft^2	+H66*@PI*H11/12	
68	PRESSURE	psia	+H90	
69	STEAM TEMPERATURE	deg F	+H102	
70	AVERAGE WALL TEMP	deg F	(+H15+H16)/2	
71	DENSITY	#/ft^3	+H101	
72	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H69+2.53E-08*H69^2	
73	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H69+0.006	
74	SPECIFIC HEAT	Btu/#-F	+H76*H73/H72	
75	LATENT HEAT	Btu/lbm	1062.3275+35.1711+(0.41242-1.023)*H69+0.00022*H69^2-9.517E-07*H69^3	
76	PRANDTL NUMBER			0.95
77	CONDENSING HTC	Btu/ft^2-hr-	+G77	
78	EFFECTIVE DELTA T	deg F	+H70-H69	
79	HEAT TRANSFER RATE	Btu/hr	@MIN(+H77*H67*H78,0)	
80	HEAT TRANSFERRED	Btu	+H79*H3/3600	
81	CUMUL HEAT TRANSFER	Btu	+H80+G81	
82	EVAPORATION	#	+H80/H75	
83				
84	BOILER THERMODYNAMCIS:			
85	The boiler is assumed to be at a uniform pressure and temperature. As with the			
86	regenerator, only the vapor phase is considered. The mass of the vapor phase changes			
87	with flow in/out of the regenerator and through evaporation/condensation.			
88	The wall temperature is assumed to be uniform.			
89	CROSS SECTION	ft^2	@PI*H13^2/(4*144)	
90	PRESSURE	psia	+H103	
91	SURFACE AREA	ft^2	@PI*H13*H14/12	
92	DENSITY	#/ft^3	+H101	
93	LATENT HEAT	Btu/lbm	+H75	
94	PRANDTL NUMBER			0.95
95	HEAT TRANSFERRED	Btu	+H17*H3/3600	
96	HEAT FLUX	Btu/ft^2-hr	+H17/H91	
97	CUMUL HEAT TRANSFER	Btu	+H95+G97	
98	EVAPORATION	#	+H95/H93	
99	TOTAL MASS OF STEAM	#	+G99+H98+H82	
100	TOTAL VOLUME	ft^3	+H89*H14+H66*H35	
101	STEAM DENSITY	#/ft^3	(@MAX(+H99/H100,0.0001)+H101)/2	
102	SATURATION TEMPERATU	deg F	453.8761+88.4691*@LN(H101)+4.6261*@LN(H101)^2	
103	SATURATION PRESSURE		(0.496-0.023*H102+0.0004241*H102^2-2.667E-06*H102^3+1.09E-08*H102^4+7.767E	

Three-Port model of Thermally Activated Pump:
Simultaneous flow through inlet and outlet.
Constant flux boiler, fixed steam mass

5	TIME	SEC		824.2980
6	TIME STEP	sec		0.006
7	DESIGN PARAMETERS:			
8	INLET ID	in		0.75
9	INLET LENGTH	ft		6
10	OUTLET ID	in		0.75
11	OUTLET LENGTH	ft		24
12	INLET TEMPERATURE	deg F		60
13	DENSITY	#/ft^3		62.4
14	REGEN ID	in		0.75
15	REGEN LENGTH	ft		20
16	BOILER ID	in		2
17	BOILER LENGTH	ft		1
18	COLD WALL TEMPERATUR	deg F		70
19	HOT WALL TEMPERATURE	deg F		70
20	HEAT TRANSFER RATE	Btu/hr		30000
21				
22				
23	OUTLET COLUMN:			
24	CROSS-SECTION	ft^2	@PI*H10^2/(4*144)	
25	VELOCITY	ft/sec	(H59*H57-H41*H36)/H24	
26	ACCELERATION	ft^2/sec	(H60*H57-H40*H36)/H24	
27	PRESSURE OUT (P6)	psig		0
28	PRESSURE IN (P5)	psig	(+H27+H33+H13*H11*H26/(32.2*144)+4*H28)/5	
29	STAGN. PRES (P5')	psig	+H28+H13*G25^2/(2*32.2*144)	
30	FLOW RATE	GPM	+H25*H24*60*7.48	
31	REYNOLDS NUMBER		(3600*H13*H25*H10/(12*0.74))*@ABS(H25)/H25	
32	FRICTION FACTOR		0.046/H31^0.2	
33	FRICTION LOSS		4*H32*(H11*12/H10)*H13*H25*@ABS(H25)/(32.2*2*144)	
34				
35	REGENERATOR DYNAMICS:			
36	CROSS-SECTION	ft^2	@PI*H14^2/(4*144)	
37	FLOW DIRECTION		@IF(G41>0,"IN","OUT")	
38	FRICTION FACTOR		0.046/H78^0.2	
39	FRICTION LOSS	psid	4*H38*(H42*12/H14)*(H76^2/(2*H13*418000000*144))*(H41/@ABS(H41))	
40	ACCELERATION	ft^2/sec	144*32.2*(H52-H44-H39)/(H13*G42)	
41	VELOCITY	ft/sec	+G41+H40*H6	
42	INTERFACE LENGTH	ft	+G42+H41*H6	
43	DISPLACEMENT	ft	+H15-H42	
44	STEAM PRESSURE (P4)	psig	+H104-14.696	
45	FLOW RATE	GPM	+H41*H36*60*7.48	
46				
47	MIXER:			
48	The three way mixing is assumed to occur in a tee having the same run diameter			
49	as the regenerator. The regenerator and outlet are connected to the run of the tee;			
50	the inlet is connnected via the branch.			
51	OUTLET PRESSURE (P5)		+H28	
52	REGEN PRESSURE (P3)		+H51+H13*((G25*H24/H36)^2-G41^2)/(32.2*144)	
53	BRANCH PRESSURE (P1)		(@MAX(H51,H52)+4*H53)/5	
54	FLOW ERROR	GPM	+H45-H64+H30	
55				
56	INLET COLUMN:			
57	CROSS-SECTION	ft^2	@PI*H8^2/(4*144)	
58	ACCELERATION	ft^2/sec	(144*32.2*(H62-H63-H67)/(H13*H9)+H58)/2	
59	VELOCITY	ft/sec	@MAX(G59+H58*H6,0)	
60	ACCELERATION	ft^2/sec	@IF(H59>0,H58,0)	
61	STAGN. PRES IN (P0')	psig		0
62	PRESSURE IN (P0)	psig	+H61-H13*G59^2/(2*32.2*144)	
63	PRESSURE OUT (P1)	psig	(+H53+H63)/2	
64	FLOW RATE	GPM	+H59*H57*60*7.48	

65	REYNOLDS NUMBER		@MAX(3600*H13*H59*H8/(12*0.74),1)	
66	FRICTION FACTOR		0.046/H65^0.2	
67	FRICTION LOSS		@IF(H59>0,4*H66*(H9*12/H8)*H13*H59^2/(32.2*2*144),0)	
68	The water column in the regenerator is assumed to have no evaporation/condensation			
69	occurring at the liquid/vapor interface. The wall temperature and water temperature			
70	distributions are assumed to be linear, with uniform Q/A (i.e., uniform			
71	delta T).			
72	VISCOSITY	#/hr-ft		0.74
73	THERMAL CONDUCTIVITY	Btu/hr-ft-F		0.394
74	SPECIFIC HEAT	Btu/#-F		1
75	PRANDTL NUMBER		+H72*H74/H73	
76	MASS VELOCITY	#/ft^2-hr	@ABS(+H13*H41*3600)	
77	FLOW RATE	#/hr	+H76*H36	
78	REYNOLDS NUMBER		+H76*H14/(12*H72)	
79	HEAT TRANSFER COEF.	Btu/hr-ft^2-	@IF(H78>2300,0.023*(H73*12/H14)*H78^0.8*H75^0.4,4.36*H73*12/H14)	
80	WATER SURFACE AREA	ft^2	@PI*H14*H42/12	
81	WATER MASS	#	+H36*H42*H13	
82	NTU		+H79*H80/(H77*H74)	
83	WALL TEMP @ INTERFACE	deg F	+H18+(H19-H18)*H42/H15	
84	AVERAGE WALL TEMP	deg F	(H83+H18)/2	
85	COLD WATER TEMP	deg F	+H18-H87	
86	INTERFACE WATER TEMP	deg F	+H83-H87	
87	AVERAGE DELTA T	deg F	+H84-H93	
88	HEAT TRANSFER RATE	Btu/hr	+H79*H80*H87	
89	HEAT TRANSFERRED	Btu	+H88*H6/3600	
90	CUMUL HEAT TRANSFER	Btu	+G90+H89	
91	ENTHALPY FLUX	Btu	@IF(H37="IN",(H81-G81)*H74*H12,(H81-G81)*H74*H85)	
92	TOTAL ENERGY	Btu	+G92+H91+H89	
93	AVERAGE TEMPERATURE	deg F	+H92/(H74*H81)	
94	REGENERATOR THERMODYNAMICS -			
95	The steam column in the regenerator is			
96	assumed to be at a uniform pressure and at the same temperature as the steam in			
97	the boiler. Only the vapor phase is considered: The dynamics/thermodynamics of			
98	the condensed water is ignored. Thus, the steam phase is a open control volume			
99	whose mass is changing via flow in/out of the regenerator and in/out of the			
100	condensed water phase.			
101	INTERFACE VELOCITY	ft/sec	+H41	
102	BOILING LENGTH	ft	@MAX(+H15-H42,0)	
103	SURFACE AREA	ft^2	+H102*@PI*H14/12	
104	PRESSURE	psia	+H135	
105	STEAM TEMPERATURE	deg F	+H166	
106	AVERAGE WALL TEMP	deg F	(H19+H83)/2	
107	DENSITY	#/ft^3	+H169	
108	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H105+2.53E-08*H105^2	
109	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H105+0.006	
110	SPECIFIC HEAT	Btu/#-F	+H112*H109/H108	
111	LATENT HEAT	Btu/lbm	1062.3275+35.1711+(0.41242-1.023)*H105+0.00022*H105^2-9.517E-07*H105^3	
112	PRANDTL NUMBER			0.95
113	MASS VELOCITY AT INTER	#/hr-ft^2	@ABS(+H41*H107*3600)	
114	MASS VELOCITY AT EXIT	#/hr-ft^2	+H113*(H17)/(H15+H17-H42)	
115	AVERAGE MASS VELOCITY	#/hr-ft^2	(H113+H114)/2	
116	REYNOLDS NUMBER		+H115*H14/(12*H108)	
117	CONVECTIVE HTC	Btu/ft^2-hr-	@IF(H116>2300,0.023*(12*H109/H14)*H116^0.8,4.36*12*H109/H14)	
118	CONDENSING HTC	Btu/ft^2-hr-	+G118	
119	BOILING HTC	Btu/ft^2-hr-	+G119	
120	BOIL/COND/CONV?		@IF(H106>H105+10,"BOIL",@IF(H106>=H105,"CONV",@IF(H117*(H105-H106)>H118,"	
121	HTC	Btu/ft^2-hr-	(+H117+H119*(1-@EXP(-(H122/10)^20))+9*H121)/10	
122	EFFECTIVE DELTA T	deg F	+H106-H105	
123	HEAT TRANSFER RATE	Btu/hr	+H121*H103*H122	
124	HEAT TRANSFERRED	Btu	+H123*H6/3600	
125	CUMUL HEAT TRANSFER	Btu	+H124+G125	
126	EVAPORATION	#	+H124/H111	
127	BOILER THERMODYNAMCIS:			
128	The boiler is assumed to be at a uniform pressure and temperature. As with the			

129	regenerator, only the vapor phase is considered. The mass of the vapor phase changes			
130	with flow in/out of the regenerator and through evaporation/condensation.			
131	The wall temperature is assumed to be uniform.			
132	CROSS SECTION	ft^2	@PI*H16^2/(4*144)	
133	INLET VELOCITY	ft/sec	(H101*H114/H113)*(H36/H132)	
134	INLET MASS VELOCITY	#/ft^2-hr	@ABS(H133)*H107*3600	
135	PRESSURE	psia	+H167	
136	LENGTH	ft	+H17-@MAX(0,(H42-H15)*H36/H132)	
137	SURFACE AREA	ft^2	@PI*H16*H136/12	
138	DENSITY	#/ft^3	+H169	
139	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H166+2.53E-08*H166	
140	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H166+0.006	
141	SPECIFIC HEAT	Btu/#-F	+H143*H140/H139	
142	LATENT HEAT	Btu/lbm	+H111	
143	PRANDTL NUMBER			0.95
144	AVERAGE MASS VELOCITY	#/hr-ft^2	+H134/2	
145	REYNOLDS NUMBER		+H144*H16/(12*H139)	
146	CONVECTIVE HTC	Btu/ft^2-hr-	@IF(H145>2300,0.023*(12*H140/H16)*H143^0.8,4.36*12*H140/H16)	
147	CONDENSING HTC	Btu/ft^2-hr-		1000
148	BOILING HTC	Btu/ft^2-hr-		1000
149	BOIL/COND/CONV?		@IF(H152>H166+10,"BOIL",@IF(H152>=H166,"CONV","COND"))	
150	HTC	Btu/ft^2-hr-	@IF(H149="COND",+H147,+H146+H148*(1-@EXP(-(H151/10)^20)))	
151	EFFECTIVE DELTA T	deg F	@MIN(+H152-H166,150)	
152	AVERAGE WALL TEMP	deg F	+H20/(H150*H137)+H166	
153				
154	CUMUL HEAT TRANSFER	Btu	+H164+G154	
155				
156	MOD: fixed heat input and mass; calculate quality			
157	TOTAL MASS OF STEAM	#	+H132*H17*0.1	
158	TOTAL VOLUME	ft^3	+H132*H136+H36*H102	
159	SPECIFIC VOLUME	ft^3/#	+H158/H157	
160	VAPOR SPEC VOL	ft^3/#	0.586*(G166+460)/G167	
161	d(P)/d(T)	psi/F	-0.023+0.0008482*G166-8E-06*G166^2+4.36E-08*G166^3+3.0668E-11*G166^4	
162	d^2(P)/d(T^2)	psi/F^2	0.0008482-1.6E-05*G166+1.308E-07*G166^2+1.227E-10*G166^3	
163	d(vol)	ft^3/#	+H159-G159	
164	HEAT TRANSFERRED	Btu	+H20*H6/3600+H124	
165	DELTA T	deg F	(+H164-H163*(G166+460)*H161*0.185)/(1+G160*(G166+460)*G168*G162*0.185)	
166	SATURATION TEMPERATU	deg F	+G166+H165	
167	SATURATION PRESSURE		(0.496-0.023*H166+0.0004241*H166^2-2.667E-06*H166^3+1.09E-08*H166^4+7.767E	
168	QUALITY	%	(H159-0.016)/(H160-0.016)	
169	STEAM DENSITY	#/ft^3	1/H159	
170	OVERALL HEAT TRANSFER	Btu	+H154+H125+H90	
171	HEAT TRANSFER RATE	Btu/hr	3600*H170/H5	

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3	TIME	SEC		2124.3093999766
4	TIME STEP	sec		0.004
5	DESIGN PARAMETERS:			
6	OUTPUT ID	in		0.75
7	OUTPUT LENGTH	ft		12
8	DISPLACER ID	in		1
9	COLD LEG LENGTH	ft		5.5
10	HOT LEG LENGTH	ft		4.5
11	BOILER ID	in		0.75
12	BOILER LENGTH	ft		6
13	CONDENSING HTC	Btu/ft^2-hr-F		2000
14	HEAT TRANSFER RATE	Btu/hr		50000
15	PRESSURE OUT (P6)	psig		2
16	DENSITY	#/ft^3		62.4
17	AVERAGE WALL TEMP	deg F		70
18				
19	OUTPUT COLUMN:			
20	CROSS-SECTION	ft^2	@PI*H6^2/(4*144)	
21	VELOCITY	ft/sec	(H49*H47+H36*H31)/H20	
22	ACCELERATION	ft/sec^2	(H48*H47+H35*H31)/H20	
23	PRESSURE IN (P5)	psig	(+H15+H28+H16*H7*H22/(32.2*144)+9*H23)/10	
24	STAGN. PRES (P5')	psig	+H23+H16*G21^2/(2*32.2*144)	
25	FLOW RATE	GPM	+H21*H20*60*7.48	
26	REYNOLDS NUMBER		(3600*H16*H21*H6/(12*0.74))*@ABS(H21)/H21	
27	FRICTION FACTOR		0.046/H26^0.2	
28	FRICTION LOSS		4*H27*(H7*12/H6)*H16*H21*@ABS(H21)/(32.2*2*144)	
29				
30	COLD LEG DYNAMICS:			
31	CROSS-SECTION	ft^2	@PI*H8^2/(4*144)	
32	REYNOLDS NUMBER		(3600*H8*H36*H16/(12*0.74))*@ABS(H36)/H36	
33	FRICTION FACTOR		0.046/H32^0.2	
34	FRICTION LOSS	psid	4*H33*((H9-H37)*12/H8)*(H16*H36*@ABS(H36)/(2*32.2*144))	
35	ACCELERATION	ft/sec^2	144*32.2*(H38-H23-H34)/(H16*(H10-G37))+32.2	
36	VELOCITY	ft/sec	+G36+H35*H4	
37	INTERFACE LENGTH	ft	+G37+H36*H4	
38	STEAM PRESSURE (P4)	psig	+H60-14.696	
39	FLOW RATE	GPM	+H36*H31*60*7.48	
40				
41	MIXER:			
42	All mixing pressures are assumed equal			
43	OUTPUT DISPLACEMENT	ft	+H37+H50	
44	FLOW ERROR	GPM	+H52+H39-H25	
45				
46	HOT LEG DYNAMICS:			
47	CROSS-SECTION	ft^2	@PI*H8^2/(4*144)	
48	ACCELERATION	ft/sec^2	(144*32.2*(H51-H23-H55)/(H16*(H10-H50))+32.2+H48)/2	
49	VELOCITY	ft/sec	+G49+H48*H4	
50	DISPLACEMENT	ft	+G50+H49*H4	
51	STEAM PRESSURE (P4)	psig	+H60-14.696	
52	FLOW RATE	GPM	+H49*H47*60*7.48	
53	REYNOLDS NUMBER		3600*H16*H49*H8/(12*0.74)*@ABS(H49)/H49	
54	FRICTION FACTOR		0.046/H53^0.2	
55	FRICTION LOSS		4*H54*((H10-H50)*12/H8)*H16*H49*@ABS(H49)/(32.2*2*144)	
56				
57	COLD LEG THERMODYNAMICS:			
58	CONDENSING LENGTH	ft	@MAX(H37,0)	
59	SURFACE AREA	ft^2	+H58*@PI*H8/12	
60	PRESSURE	psia	+H76	
61	STEAM TEMPERATURE	deg F	+H96	
62	DENSITY	#/ft^3	+H99	
63	VISCOSITY	#/hr-ft	0.02165+4.058E-05*H61+2.53E-08*H61^2	
64	THERMAL CONDUCTIVITY	Btu/ft-hr-F	3.68E-05*H61+0.006	
65	SPECIFIC HEAT	Btu/#-F	+H67*H64/H63	
66	LATENT HEAT	Btu/lbm	1062.3275+35.1711+(0.41242-1.023)*H61+0.00022*H61^2-9.517E-07*H61^3	

67	PRANDTL NUMBER			0.95
68	EFFECTIVE DELTA T	deg F	+H17-H61	
69	HEAT TRANSFER RATE	Btu/hr	+H13*H59*H68	
70	HEAT TRANSFERRED	Btu	+H69*H4/3600	
71	CUMUL HEAT TRANSFER	Btu	+H70+G71	
72	EVAPORATION	#	+H70/H66	
73				
74	HOT LEG THERMODYNAMICS			
75	CROSS SECTION	ft^2	@PI*H8^2/(4*144)	
76	PRESSURE	psia	+H97	
77	LENGTH	ft	+H50	
78	BOILER/MANIFOLD AREA	ft^2	@PI*H11*H12/12	
79	BOILER/MANIFOLD VOL	ft^#	@PI*H11^2*H12/(144*4)	
80	TOTAL SURFACE AREA	ft^2	@PI*H8*H77/12+H78	
81	DENSITY	#/ft^3	+H99	
82				
83				
84				
85	MOD: fixed heat input and mass; calculate quality			
86	TOTAL MASS OF STEAM	#	+H75*H10*0.1	
87	TOTAL VOLUME	ft^3	+H79+H77*H75+H58*H31	
88	SPECIFIC VOLUME	ft^3/#	+H87/H86	
89	VAPOR SPEC VOL	ft^3/#	0.586*(G96+460)/G97	
90	d(P)/d(T)	psi/F	-0.023+0.0008482*G96-8E-06*G96^2+4.36E-08*G96^3+3.0668E-11*G96^4	
91	d^2(P)/d(T^2)	psi/F^2	0.0008482-1.6E-05*G96+1.308E-07*G96^2+1.227E-10*G96^3	
92	d(vol)	ft^3/#	+H88-G88	
93	BOILER HEAT	Btu	+H14*H4/3600	
94	CUMUL HEAT TRANSFER	Btu	+H93+G94	
95	DELTA T	deg F	(+H93+H70-H92*(G96+460)*H90*0.185)/(1+G89*(G96+460)*G98*G91*0.185)	
96	SATURATION TEMPERATU	deg F	+G96+H95	
97	SATURATION PRESSURE		(0.496-0.023*H96+0.0004241*H96^2-2.667E-06*H96^3+1.09E-08*H96^4+7.767E-12*H96^5	
98	QUALITY	%	(H88-0.016)/(H89-0.016)	
99	STEAM DENSITY	#/ft^3	1/H88	
100	OVERALL HEAT TRANSFER	Btu	+H94+H71	
101	HEAT TRANSFER RATE	Btu/hr	3600*H100/H3	